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November, 1941

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Heating · Piping and Air Conditioning

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mercial, Industrial
Large Buildings

TO SAVE fuel for heat-
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SPECIAL!
Air Conditioning
Priorities Program
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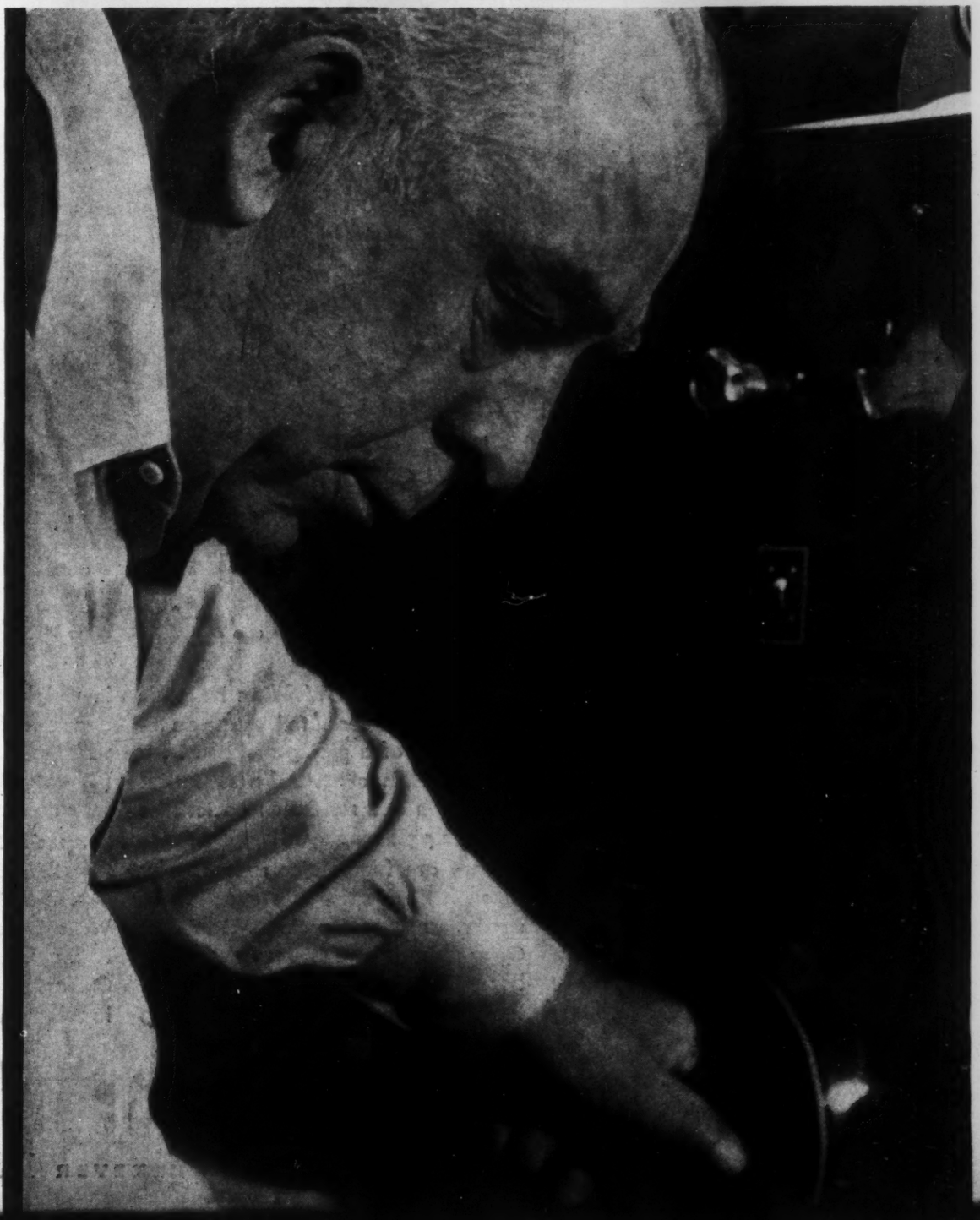

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Here are four major reasons why these fittings speed-up erection of strong, continuous piping systems: (1) Grinnell Welding Fittings weld easier, because they are *metal-matched* with the pipe itself; (2) their ends match-up in weight to standard-weight, extra-strong or O.D. pipe or tubing; (3) their welding faces are correctly scarfed; (4) only plain circumferential butt welds are required to join them.

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WELDING FITTINGS BY **GRINNELL**

WHENEVER **PIPING** IS INVOLVED

The Editor's Pages

GOVERNMENT SEEKS ENGINEERS

The United States civil service commission announced last month that the government is seeking heating, ventilating, air conditioning and plumbing engineers for national defense jobs. The salary range is from $2600 to $5600 a year. If you are qualified for and interested in getting one of these jobs, apply under the civil service examination announcement No. 69 for *engineer*. The required application forms are Nos. 8 and 4006-ABCD.

***CORK**—Because of continuing shortages of cork, limitations were placed around deliveries of insulation board containing cork, in an order issued last month by Donald M. Nelson, director of priorities. Only defense orders, and orders for food preservation purposes, may be filled with such board. An interpretation of this order, M-8-a, defined "insulation board to be used for the preservation of food" as that to be used in cabinets and other small cold storage boxes in which the temperature is to be maintained below 20 F, and cold storage rooms and walk-in boxes, in which the temperatures are to be kept below 40 F*

BOILERS AND RADIATORS STANDARDIZED TO CONSERVE

The boiler and radiator industry is simplifying and standardizing its products in order to conserve vital raw materials needed for national defense, according to the Institute of Boiler and Radiator Manufacturers.

As a result of action on the part of manufacturers, the U. S. department of commerce, through the national bureau of standards, has announced a revision of simplified practice recommendation R-174-40 for large tube cast iron radiators. This revision constitutes a reduction in the number of sizes of large tube radiators to a total of 13 sizes and became effective October 1. On this same date, all manufacturers also put into effect a program covering small tube radiators which reduced the number of standard sizes to 10. By January 1, the number of sizes of radiators will be approximately 20 instead of more than 50 which had been produced prior to 1940.

The manufacturers of radiators are asking the assistance of all buyers of radiation to accomplish this objective. Government departments, including federal, state and municipal, are being urged to discontinue specifying any radiators except those being manufactured under the new revision. Architects, engineers and contractors are asked to coöperate in this effort.

Some manufacturers are temporarily discontinuing the manufacture of convectors to conserve the steel required for enclosures and also to give the manufacturer more foundry space for the manufacture of cast iron radiators.

The problem of reducing the number of boilers is somewhat more complicated than the radiator situation, but action which is currently under way on the part of each of the individual manufacturers will result, within the next few months, in a substantial reduction in the variety of cast iron boilers which will be introduced. The program will be carried out in such a way that the needs of the building industry will be fully served by the variety of boilers which will still be available, it is believed.

***ARC WELDING SCHOLARSHIPS**—Recipients of the arc welding scholarships in welding engineering established at Ohio State University in 1939 are William S. Wagner and Robert Frederick Bluhm. These scholarships, made possible by an anonymous donor, amount to $250 each and may be awarded annually to one junior and one senior student in the curriculum in welding engineering*

DEFENSE CONSTRUCTION TO BE DISCUSSED

According to the Chamber of Commerce of the United States, a grand total of approximately $7½ billion has been authorized for industrial plant construction, aeronautical and other plant facilities, military housing and cantonment facilities and defense housing and community facilities. As of September 1, a little more than $3 billion of

COPPER USE RESTRICTED DRASTICALLY

Conservation order M-9-c of OPM's priorities division issued late last month curtails the use of copper and copper base alloys in the manufacture of a long list of products where such use is not absolutely necessary for defense or essential civilian requirements. Included in the list of items for which use is prohibited are air conditioning equipment (except small, moving parts and bearings); conduits and tubing; grilles; bands on pipe covering; convectors and local heaters; hot water heaters, tanks, and coils (except as provided in the defense housing critical list issued September 12); pipe and fittings (except as provided in the defense housing critical list); ventilators and skylights; water containers for humidification; weatherstripping and insulation; fire hose, couplings and fittings (except brass expansion joints and valve trim); and valve handles.

No person may use or apply copper or copper base alloy sheet, strip or screening for any purpose in connection with building construction except for minor repairs or maintenance (but not replacement) on existing buildings where the existing installation is copper or copper base alloy and no substitute can be used.

The restrictions do not apply to copper and copper base alloys for several uses, among which are use in chemical plants to the extent that corrosive action makes the use of any other material impractical; and for use as condenser tubes, heat exchanger tubes (other than radiators which are used to control air temperature) and tube sheets in oil refining plants, in plants generating steam for public or industrial use and in plants generating electric power for public or industrial use, where and to the extent that corrosive action makes the use of any other material impractical.

CONSTRUCTION ACTIVITIES OF WAR DEPARTMENT

S. 1884, a bill to transfer all construction activities of the war department to the corps of engineers, was introduced at the instance of Robert P. Patterson, undersecretary of war, with the approval of the chief of staff of the army, the secretary of war, and President Roosevelt. In a statement made before the senate military affairs committee in support of the bill, Mr. Patterson said that the advantages of having the entire construction program in charge of one branch are too manifest to require detailed discussion; with the work divided between two branches, there must necessarily be a lack of coördination, he stated, and also a duplication of effort. He expressed belief that a gain in efficiency and in economy would result from concentrating all construction work in one branch and holding that branch accountable for results.

Granting that the entire construction program should be in charge of one branch of the war department, the remaining question is which branch should be entrusted with the task. The possible solutions would be to place the work with the engineers, with the quartermaster corps, or with a new branch charged solely with construction. The engineer corps is the logical one to select, he said, because in normal times it has a well established, large, active organization for construction work due to the many projects of a civilian character in which it is engaged. In time of war or national emergency it requires no great effort to turn that organization to the task of building what may be needed for the army. The quartermaster corps, on the other hand, has little to do in the way of construction in normal times, and its organization is necessarily not on a large scale. Thus, with the coming of an emergency, it has to build its organization from the grass roots. Also, other activities of the corps of engineers are more closely related to construction work than are the other activities of the quartermaster corps, one of the prime functions of the engineers being to carry on construction and engineering in an active theater of operations.

As to a separate branch devoted solely to construction work, as recommended by the report of the Truman committee on camp construction issued last August, Mr. Patterson expressed the belief that such a separate branch would suffer from neglect in ordinary times because of lack of substantial work to do and would then be ill prepared to carry on a large program in case of sudden emergency.

Under the bill, the work would be done by civilian contractors as before, the difference being that the contracts would be made by the corps of engineers and construction would be under supervision of that corps. The bill also provides that maintenance and repair of buildings be placed with the corps of engineers as well as the responsibility for acquisition of real estate.

the program had been completed and the current program for 1942 amounts to over $2½ billion. Unofficially, however, it is expected that the 1942 program will be doubled.

Problems raised by this tremendous defense construction program are to come under the scrutiny of the fifth annual construction industry conference, to convene in Washington on November 6 for a two day session.

SMA REPORTS 758,492 STOKER INSTALLATIONS

In coöperation with and at the specific request of government defense officials, the Stoker Manufacturers' Association, 307 N. Michigan Ave., Chicago, Ill., has made a number of economic and statistical surveys and studies indicating the importance of stokers in the national defense program, especially with respect to the conservation of fuel.

Through the coöperation of stoker manufacturers representing over 90 per cent of the total unit volume of the industry for installations made during the past 15 years, the association is publishing a statement showing a total of 758,492 stoker installations.

AIR CONDITIONED BLACKOUT PLANT

Completion of the new $9,066,000 supercharger plant of the Allis-Chalmers Mfg. Co. in about six months will give Milwaukee its first real blackout factory. Soon after that the 500 x 800 ft single floor building in suburban Greenfield township will be ready to house modern production facilities for turning out the urgently needed mechanism that, along with the U. S. bombsight, has been described as potentially one of the most decisive weapons of the war.

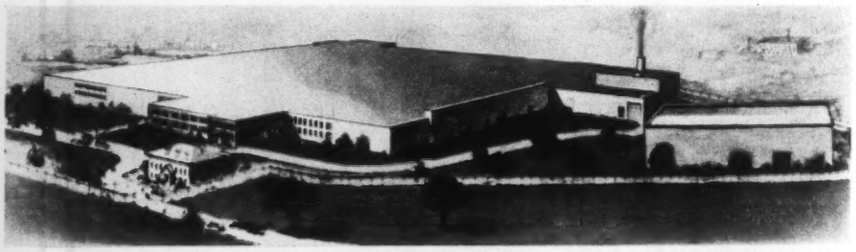
Built entirely around planned production operations, the new factory is to be of purely functional design. According to Carl E. Meyer, head of the Allis-Chalmers architectural staff, the single purpose of production for immediate defense needs has been constantly kept in mind.

Most modern provisions, however, will not only contribute to production efficiency, but provide ideal working conditions as well. Air conditioning, for example, will control temperature and humidity throughout. There will be four refrigerating units having a combined capacity of 2650 tons.

In the large test rooms, where all finished materials will undergo strictest examination, complete acoustical control will be provided, while the entire building will be specially insulated. Other facilities will include a fully equipped laboratory for research and testing of raw materials plus modern shower and locker rooms.

Only the 50 x 200 ft area devoted to offices and rest rooms will have windows. Fluorescent lighting for the entire area given to production will be designed to allow normal operation of the plant through any air raid attacks that may ever affect this inland location. Additional protection against such military eventualities results from location of vital power lines underground, the electrical sub-station to be completely enclosed and protected against the effect of possible bombing.

Adjacent to the main building will be a forge plant, 80 x 200 ft, as well as a terminal building already constructed.



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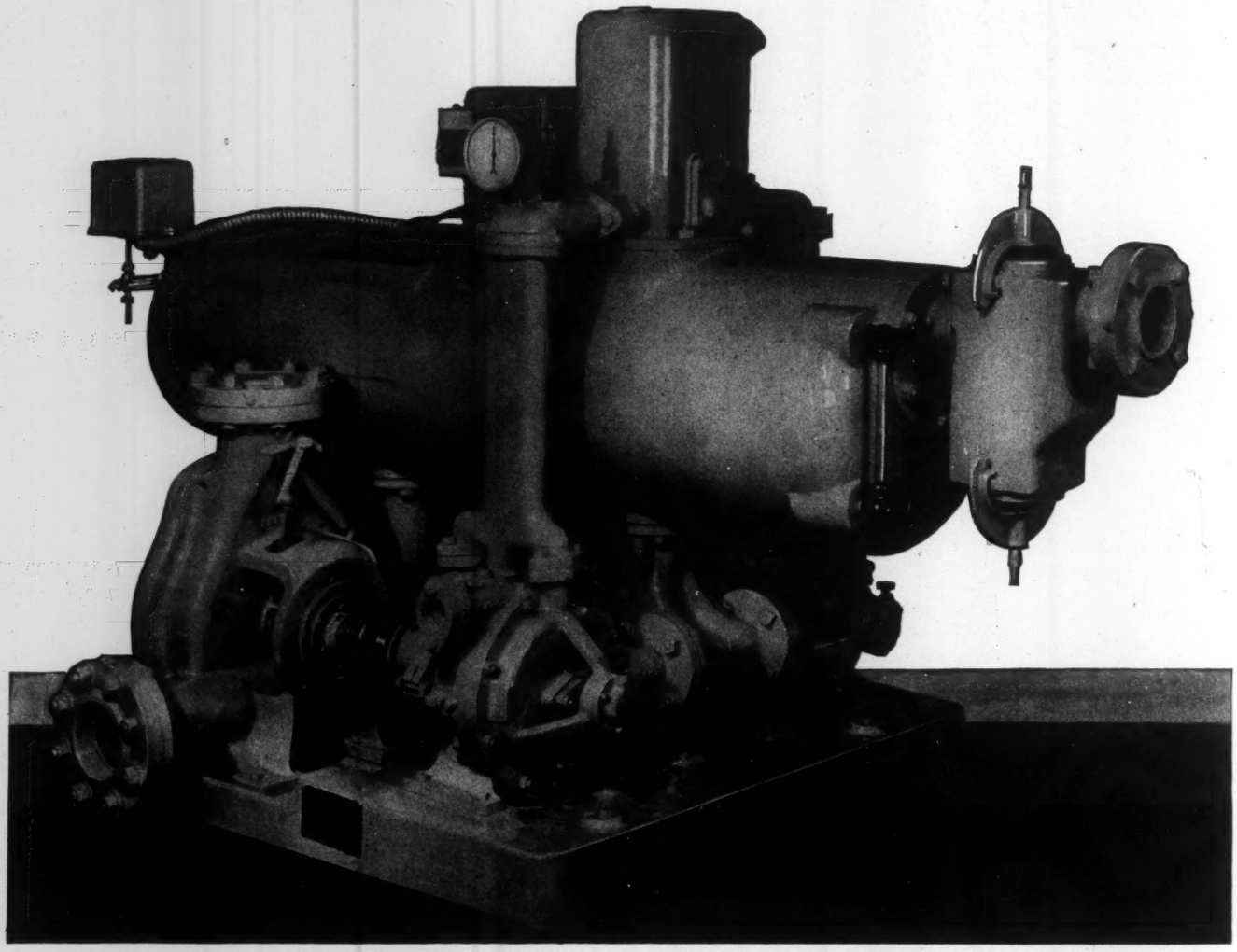
Where building heat *MUST NOT FAIL* always specify the Nash Vapor Turbine



THIS SAFE NASH HEATING PUMP DOES NOT REQUIRE ELECTRIC CURRENT BECAUSE IT OPERATES ON THE SAME STEAM THAT HEATS THE SYSTEM

In Hospitals, Greenhouses, Schools, Public Buildings, Theatres, wherever heating systems must not fail, install the Nash Vapor Turbine, for it is entirely independent of electric current failure, and continues to operate as long as there is steam in the system.

More than that, the Vapor Turbine is a most economical pump, for the elimination of electric current does away with current cost, the largest single item in the operation of an ordinary return line heating pump.

Greater savings still are effected by the Vapor Turbine in the system, for the reason that this pump operates continuously. It is the only pump that can do this with economy. Continuous operation means uniform circulation, and uniform circulation saves steam.

The Nash Vapor Turbine has but one moving part, rotating in the casing without metallic contact, and requiring no internal lubrication. Quiet, compact, and trouble-proof. Bulletin A-290 is free on request.

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Conservation for National Defense

How to Cut Heating Cost

E. A. Tyler, Rental Manager, Tells of Steps Taken to Save Over 40 Per Cent of His Building's Steam

HOW TO SAVE the fuel required for heating—essential service in plants, commercial buildings, institutions and all structures—is a vital problem to operating men always. During our national effort in defense production, it is even more important and becomes a patriotic duty. . . . Mr. Tyler, rental manager of the First National Bank building, St. Paul, Minn., explains in detail here the steps taken to reduce the steam consumption for heating this large office building. This conservation program has resulted in a saving of over 40 per cent during the period of the last few years

WHEN we built our new 30 story building in 1930-31 —it was really an addition to an older 16 story building—we purchased a control system for the two pipe vacuum heating installation that seemed to offer the best chance for a long life and successful operation. This system proved to be entirely satisfactory, performance being within the guarantee of its producer.

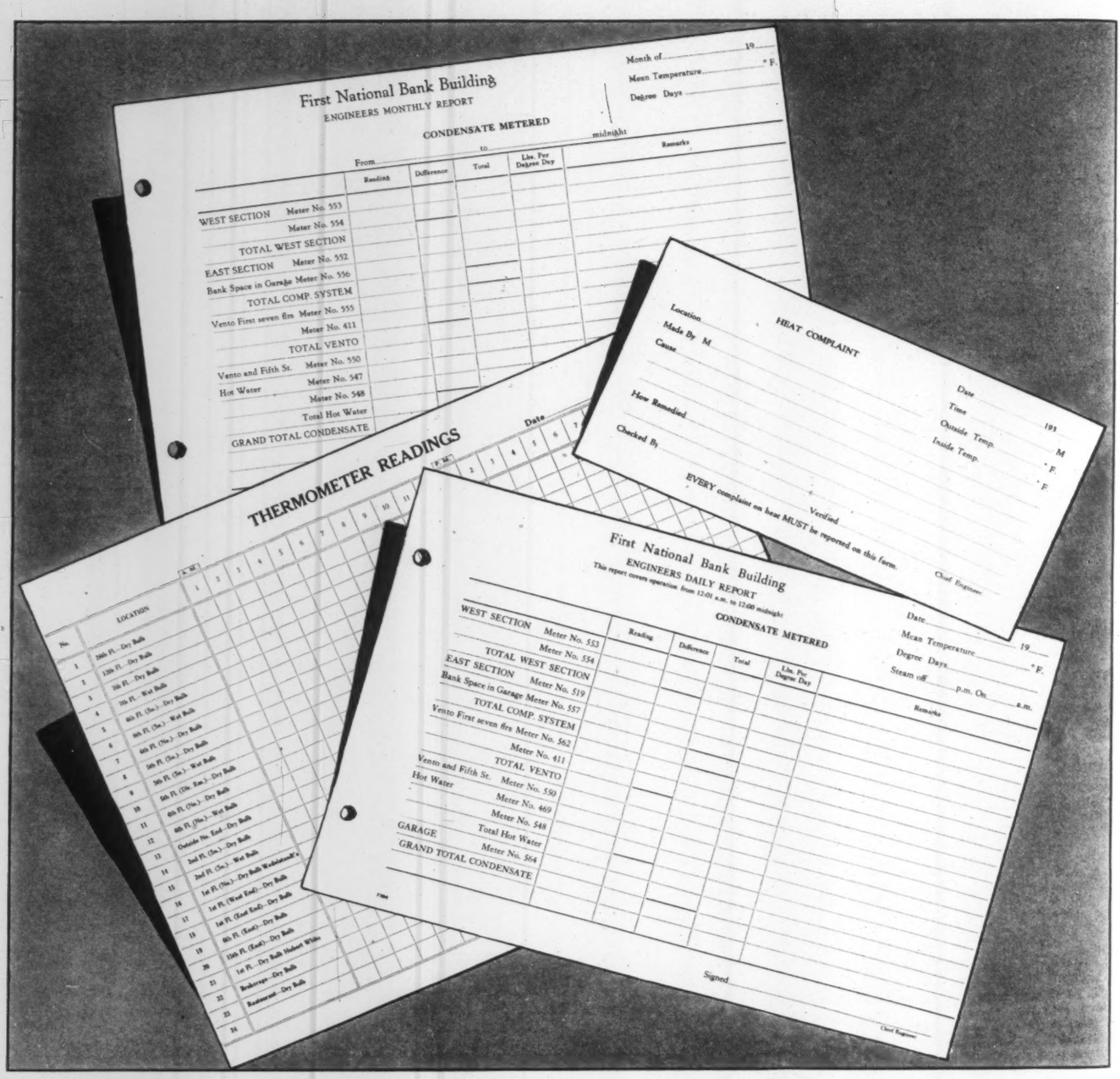
However, there were some troublesome features, and after a few years' use the original reducing valves commenced to give trouble. We turned to our able heating contractor and assigned him the job of finding a better means of control. We wanted something that was accurate, that could be manipulated with the finger tips, and was reasonable in cost.

Control Pays

Our heating contractor found a control primarily designed for accurately governing the heat in industrial ovens that seemed to be the answer to our problem. We installed one as a trial, together with a new reducing valve of the finest quality obtainable. The combination worked perfectly and we immediately proceeded to equip all zones with the new reducing valves and controls, mounting the recorders and tuning dials in the chief engineer's office. Now the man on duty can vary the pressure with no more effort than it takes to raise his arm and turn a dial. The total cost of this change was only $2000, which was more than paid for from the first year's savings.

Checking temperatures around the building, summer and winter, took about three hours per day of one man's time, and the readings were not very reliable. The chief needed an hourly check on outside temperature and inside tempera-

Controls for the five heating zones are in the chief engineer's office. Pressures can be regulated by turning the small button for each zone



These simple forms assist in efficient, economical operation

tures throughout the building, if he was to play the right tune on those dials. So we bought and installed an electric resistance thermometer with 25 locations within the building and one outside, mounting the instrument right over the chief's desk. [*See front cover.*] We also provided a form for hourly recording of temperatures. The cost of this equipment installed ran about $1,000.

The first season of operation of the new building, pressure schedules calibrated to the outside temperature were worked out, but they provided for adjustments in large jumps. With the completion of improved controls and reducing valves, and the resistance thermometer for checking inside temperatures, we felt certain that these schedules could be greatly improved. First, we devised a chart so that a curve could be drawn on to show required pressures, and after a month of experimenting in the fall and midwinter each we established permanent curves that show at a glance required pressure for any outside temperature.

These curves, however, cannot be taken literally, but are always subject to correction by the chief engineer or the man on duty, as his checking of inside temperatures indicates errors. Perhaps if one had a sun indicator and wind direction and velocity gage registering in the engineer's office, more accurate curves could be set up. I considered these instruments, but found that the cost was in excess of any possible savings and that the added complications for the operating men might be excessive. Each year, as the men have become more skilled in using these controls, has shown a steady decline in steam consumption for space heating.

Our chief engineer is a fine, loyal

man who learned his trade the hard way by shoveling coal and polishing brass. It was after the completion of the installation of the new reducing valves, controls, and thermometer that he really loosened up. I believe that it dawned on him then that we were not just "monkeying around," but were providing him with tools that he could take pride in and from which he obviously derived great personal satisfaction. From then on he has really put his heart into the whole program. Gruffly and persistently, he has trained his men to the slogan "No wasted steam,—no cold complaints from tenants." Each of the past four years has shown a decline in steam consumption, no small part of which has been due to the improvement in the skill of the men who operate the plant.

Accounting System

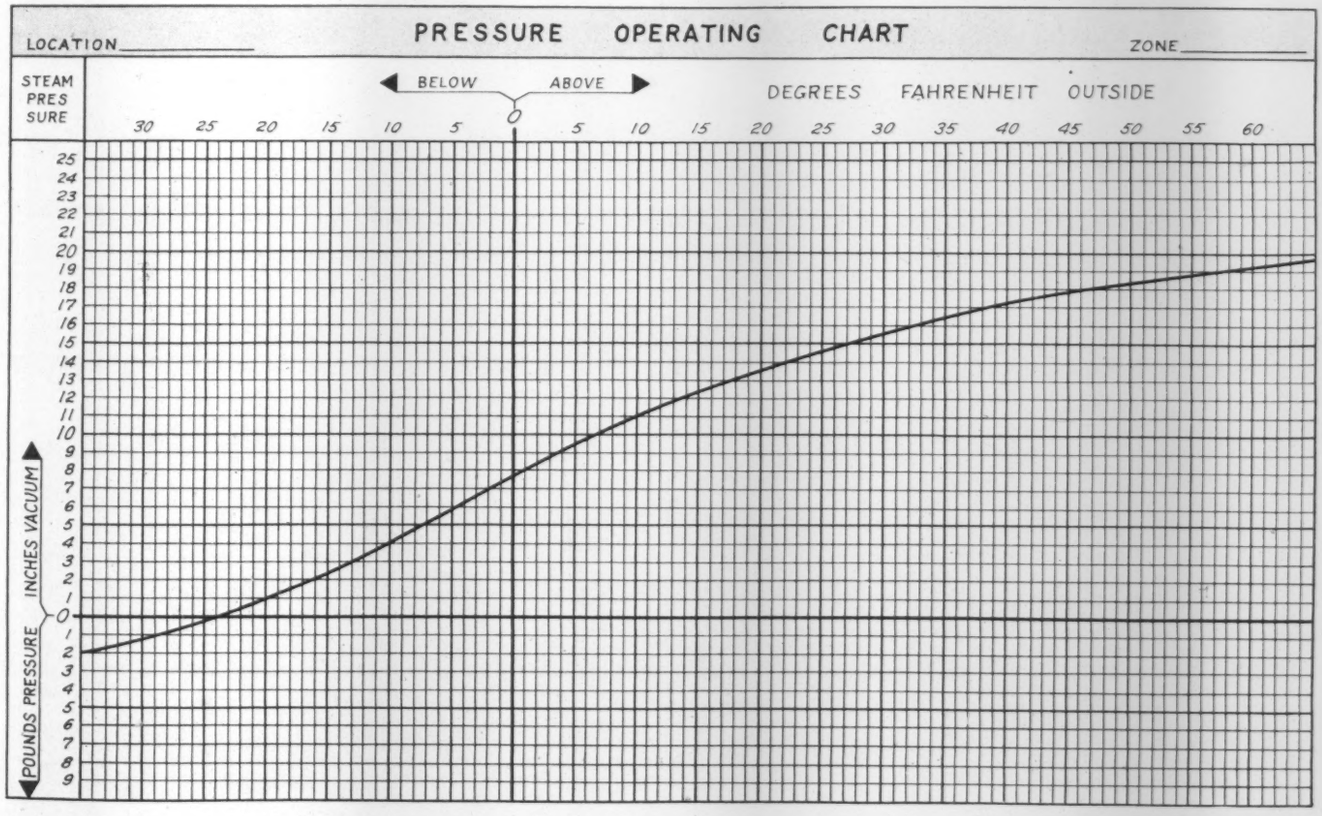
The accounting system consists of a few simple forms for recording daily and monthly steam metered on each zone. The meters are read at 12 o'clock midnight and recorded on a small tabbed form which goes to the chief engineer, who makes up his daily report in the morning. This report shows the beginning and ending numbers of each meter, total steam consumed on each zone, and steam consumed per degree day for each zone and total. It also shows mean temperature and degree days for the 24 hr period.

This report comes to me each day and at the end of the month a report showing the same data for the entire month is made up by the chief engineer and delivered to me. The power company's bills for steam are checked from these reports. I record the monthly data on a handmade form which I keep in my desk, and at the end of the heating season I summarize the entire season on another handmade form also kept in my desk. This seasonal summary shows for each zone and the total, gross consumption, degree day consumption, and consumption per thousand cubic feet per degree day for space heating for each building. By computing the last named figure we are able to compare results with any other building that keeps similar records. The only other forms used are tabbed forms for recording the hourly temperatures within the building and outside, and a form for recording cold complaints from tenants. We also maintain a recording thermometer from which mean temperatures are computed.

From these daily reports it is easy to prepare graphs showing degree days and steam consumption per degree day for each zone and total. During the first two or three years I maintained running graphs on these items, but as I was able to memorize efficient norms for each zone and total, the graphs were no longer useful, as I could tell from a glance at a daily or monthly report if the efficiency was being maintained.

The Results

When we built the new building we made a 10 year contract with the power company to furnish us steam. This contract was based on a calculated consumption of 61,086,000 lb per heating season of 7500 degree days, which total was arrived at by the engineers for the power company, and which was based on actual consumption of a building a block away of similar size and occupancy. This estimated seasonal consumption in-


LOCATION
PRESSURE OPERATING CHART
ZONE
STEAM PRESSURE
BELOW
ABOVE
DEGREES FAHRENHEIT OUTSIDE
30 25 20 15 10 5 0 5 10 15 20 25 30 35 40 45 50 55 60
25 24 23 22 21 20 19 18 17 16 15 14 13 12 11 10 9 8 7 6 5 4 3 2 1 0 1 2 3 4 5 6 7 8 9
POUNDS PRESSURE
INCHES VACUUM

cluded steam for all purposes for the 7,500,000 cu ft in the old and new buildings, a 300 car garage and a two story building 100 ft by 130 ft with a connecting arcade. The power company's estimate was checked by our own engineers, who advised that it was a close estimate and might be exceeded.

Consumption for space heating for 1932-33 was 43,035,000 lb. This figure was very gratifying and fully justified our purchase of the original control system during the construction period, and we proceeded in a self-satisfied manner to forget the matter. That was a grave error, for during the next two seasons consumption increased sharply and it was not until December of the following year that we started to work on our experiments and improvements. Table 1 shows steam consumption per degree day except for hot water heating.

Total consumption for all purposes for the 1940-41 season was 34,790,700 lb with 7244 degree days. Adjusted to 7500 degree days, total consumption would be 36,000,000 lb, which (compared to 61,086,000, the original calculated requirement) shows a saving of 41 per cent. I realize that the engineers might have erred in their original calculations, but I am inclined to believe from my experience with engineers that the error, if any, would be on the near, rather than the far side, as their purpose was to sell us on purchased steam. Furthermore, you can find plenty of large buildings today that are consuming steam on the basis of the engineers' original calculations.

Table 1—Steam Consumption for Space Heating

Season	32-33	33-34	34-35	35-36	36-37	37-38	38-39	39-40	40-41
Lb Steam per Degree Day	5500	6240	6042	5064*	5070	5150	4952**	4582	4391

* Commenced experiments and improvements.
** New controls and thermometer installed.

The real key figure is the consumption per 1000 cu ft per degree day for space heating. For our last heating season (1940-41) this figure was 0.407, and I believe that we can reduce consumption another million pounds or more this coming season as the secondary dampers referred to in my article in the August HPAC were not installed until last January.

More Comfort

I have talked so much about saving steam that you might have the impression we reduced the comfort level. As a matter of fact, we improved it.

We are now maintaining a much more uniform temperature throughout the buildings than we did before the improvements were installed, and cold complaints from tenants have been materially reduced. Some rooms that we never could heat satisfactorily are now comfortably heated. The north arcade was cold and at temperatures of 20 F and below, the draft from this arcade cooled off our ground floor, elevator shafts, and savings department. We cured this by isolating the condensate from the indirect heating coil which is returned at about 180 F, pumping it through a blast coil and recirculating 2000 cfm of air through this coil into the cold arcade. We did not save any steam on this directly, but we cured a bad situation at a cost of the pump, coil, piping and ducts; and the maintenance cost of a 3 hp motor.

We were exhausting about 10,000 cfm of warm air in the winter from the main banking room, and operating a 300 car garage adjoining our main building, which had to be heated. On a visit to Montreal we picked up the idea of turning this warm air into the garage to offset some of the heating load. This was done by connecting the exhaust openings by an insulated duct over the roof of a low part of the building, to the outside air ducts in the garage. This saved almost 50 per cent of the steam needed for heating the garage.

We did not have defense in mind when we started these improvements, but we are glad now that we can look Uncle Sam in the eye and say, "We are not wasting fuel."

Coordinate Dimensions of Building Equipment

Under the sponsorship of the American Institute of Architects and the Producers' Council, the American Standards Association organized in September, 1939, a representative sectional committee to undertake project A62, having for its objective and scope:

a) The development of a basis for the coördination of dimensions of building materials and equipment, and the correlation of building plans and details with such dimensions.

b) Recommendation of sizes and dimensions as standards suitable for dimensional correlation.

The advantages of a greater degree of size standardization, through the coördination of dimensions of building materials and equipment, have long been recognized by those seeking economies in the fabrication of the component parts of buildings and their assembly in completed structures.

While individual groups producing certain materials and items of equipment have, from time to time, effected improvements and simplifications in the standardization of their particular products, it has long been evident the most effective interrelationship between the dimensions of all materials and assemblies forming the completed structure can only be attained by a comprehensive and coördinated study of all factors.

The membership of sectional committee A62 includes representatives of 38 professional and industrial associations and organizations and eight members-at-large, with Max H. Foley, architect, as chairman.

The executive committee of ASA project A62 has recently published a brochure of 65 pages containing information concerning the organization, scope, purpose and advantages of the project with detailed layouts illustrating the application of the coördination of dimensions to the plan and elevation of a housing project, and with numerous examples of the detail of application to various forms of construction, wood or metal windows, wood doors, height coördination, joist and slab floor construction, etc. This brochure has been priced at $1.00; it may be ordered from M. W. Adams, at 110 Arlington St., Boston, Mass.

Production for National Defense

Modern Warfare Has Made the Air Conditioning of Plants Essential

MODERN war demands fighting machines in hitherto unknown numbers. For a nation to survive, its production of war machines must be uninterrupted. To conceal its essential centers of production from night raiders of the air, its factory buildings must be of the "windowless" or, at any rate, of the "blackout" variety, tightly sealed against escape of any chance ray of light.

Of necessity, factories are filled with heat producing machinery and processes. The sun pounds untold millions of heat units down through their vast roof expanses. Without the openings customary in the past, this heat would result in unbearable inside temperature in the summer. Thus, modern war has made industrial plant air conditioning of one form or another a defense necessity. One of the most recent examples is the new Packard Motor Car Co. plant, Detroit, for the production of Rolls Royce aircraft engines.

To speed production without sacrifice of precision and quality, manufacturing of the Rolls Royce Merlin engine for the RAF will be carried out at this new plant in an atmosphere of 75 F dry bulb temperature and 50 per cent relative humidity even in midsummer weather. Both the aircraft assembly building and the test and teardown building are air conditioned, the "indirect" method being used. The cooling effect is distributed to the various fan systems by means of cold water chilled by two 500 ton centrifugal refrigeration machines.

There are four fan systems in the assembly building alone, each circulating 55,000 cfm of air through a system of overhead ducts. The systems are arranged to introduce varying proportions of outside air between 40 and 100 per cent of the total air circulated. The same systems are used for winter heating.

The equipment for the various air handling systems is situated in penthouses at the roof level. Each of the systems contains the following major items: Two outside air dampers, one for handling 40 per cent and the other for handling 60 per cent of the total air; one recirculating damper; one relief damper; one set of removable, cleanable air filters; one set of tempering coils for heating service with steam; one set of chilled water air conditioning coils; one set of reheating coils for heating service with steam; one winter humidifier; one set of entrained moisture eliminator plates; one centrifugal fan with capacity of 55,000 cfm at 1¾ in. static; one three phase, 60 cycle, 440 volt, 1200 rpm motor with v belt drive; and the necessary controls and accessories.

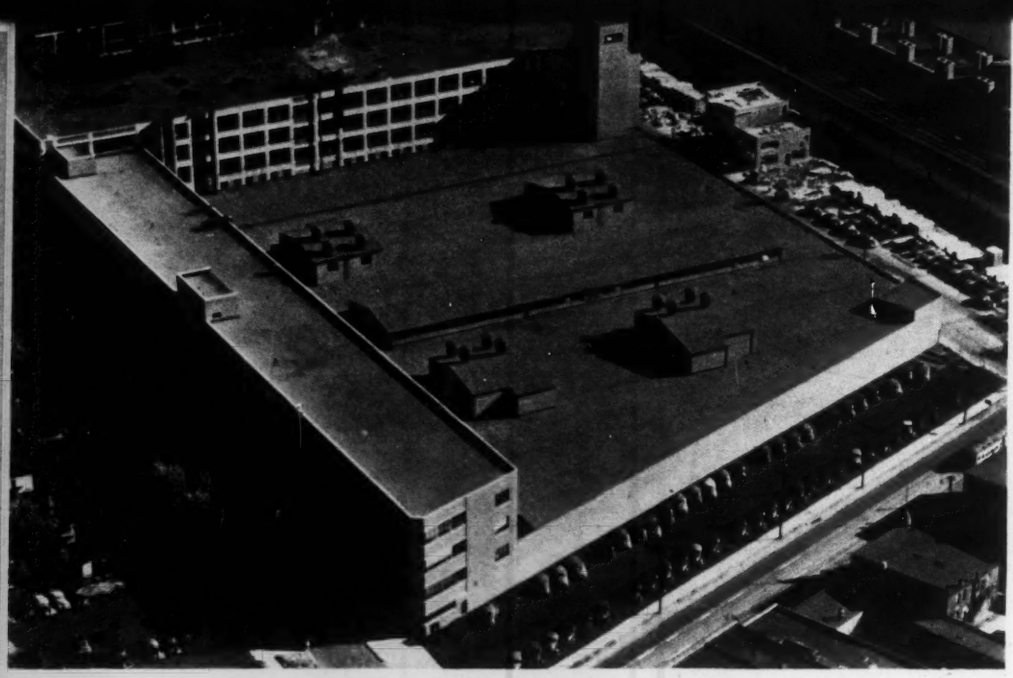
MODERN WARFARE—demanding as it does fighting machines in heretofore unknown numbers, and employing air attacks on the industrial facilities of the enemy—has made the air conditioning of windowless, blackout plants essential to efficiency and uninterrupted operation. Thousands of tons of refrigeration for air conditioning serve America's new defense plants. . . . At the Packard Motor Car Co.'s new factory for the production of Rolls Royce Merlin aircraft engines for the RAF, two 500 ton centrifugal refrigeration units supply chilled water to the four fan systems for the assembly building. The air handling equipment, to which the chilled water is piped, is installed in four penthouses on the building's roof. . . . This description was prepared by F. O. Jordan, consulting engineer. The building was not engineered by him, however. The architects and engineers were H. E. Beyster Corp.

Operation of Systems

The refrigerating capacity of the centrifugal compressors is varied manually by regulation of suction throttling valves or dampers. Adjusting these valves toward the closed position reduces the refrigerating capacity of the machines because the pressure and hence the temperature of the evaporators in the water coolers is raised, and because the pressure (and hence the density) of the gas to the compressors is reduced so that less gas is handled.

The controls for the remainder of the system are fully automatic. Even the change-over from heating to cooling is accomplished without attention. The control system is pneumatic with motor operated valves and dampers.

As long as the fan is out of operation, the outside air and relief



Air conditioned, indirectly lighted main assembly building, showing the four penthouses for the air handling equipment

dampers are closed. As soon as the fan is placed in operation, the 40 per cent outside air damper opens. As long as heat is required in the building, a thermostat in the return air duct holds the 60 per cent outside damper closed.

If the mixture temperature of the outside and recirculated air falls below 50 F a thermostat in the mixture chamber closes the 40 per cent damper.

In addition to controlling the position of the 60 per cent damper, the return air thermostat also modulates steam to the reheating and tempering coils in sequence to maintain the desired dry bulb temperature in the building. If the outside air temperature falls to 32 F, the outside air thermostat holds the steam valve open to the tempering coil, while the return air thermostat modulates the steam valve to the reheater coil to maintain the desired building temperature. A humidistat in the return duct controls the humidifier to maintain 30 per cent relative humidity in the building.

In "between season" weather, when heating or refrigeration are unnecessary, and when the mixture temperature of outside and recirculated air is above 50 F, the 60 per cent damper is permitted to open to introduce outside air in the quantity required for removing internal heat.

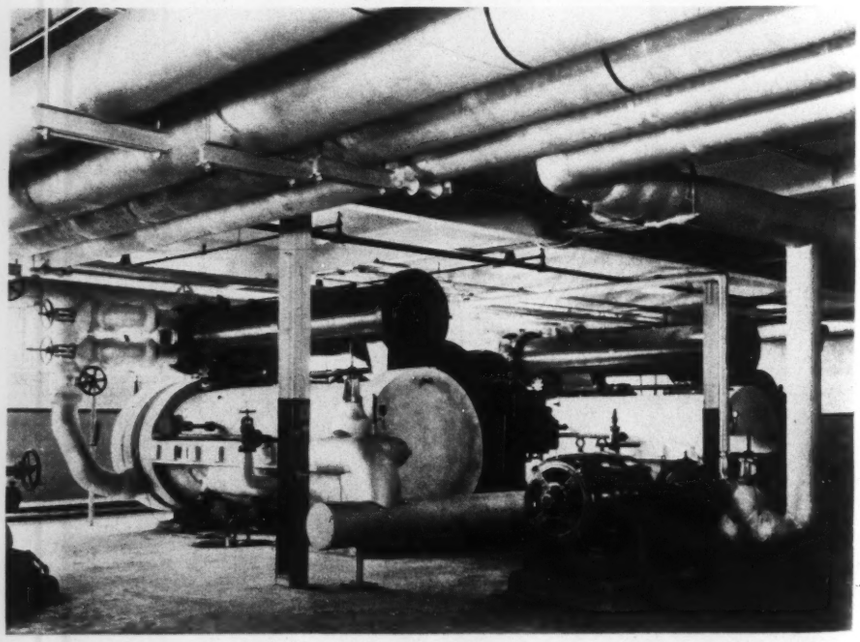
When the outside temperature rises to a predetermined point, a thermostat in the outside air intake automatically changes over the control system to summer operation. The thermostat and humidistat in the return air duct are reversed and reset from the winter setting of 72 F and 30 per cent to the summer setting of 75 F and 50 per cent. The humidifier is cut out of service, and the 60 per cent damper is modulated toward the closed position with rising outside wet bulb temperature.

Under summer operation, the chilled water supply to the indirect air conditioning coil is regulated by means of a three way valve operating under control of both the thermostat and the humidistat in the return air duct to prevent rise of building temperature and humidity above the desired effective temperature. Under conditions when the sensible load on the building is so light in proportion to the latent load that overcooling would result, the reheating coil comes into service.

In the aircraft engine industry there has always been one slogan: Quality must never give way to production; quality must be maintained at all times, and production must be worked out based upon the necessity for quality.

As the British handled it, the Rolls Royce was practically a handmade engine; even under the terrible beating of repeated bombing raids, the Rolls Royce never was produced by methods of mass production.

The British have made the Merlin the hard way. In America the hard way will be made not easy, but easier. — From a radio address by Major-general George H. Brett on the occasion of the Packard Motor Car Co.'s first production of the Rolls Royce engine.



Part of the air conditioning equipment serving this important defense factory

CHECK WEAK SPOTS **IN PIPING LAYOUT --to avoid shutdowns**

By A. C. Kirkwood

"See what you need, and we'll arrange for the time to do the work. Suppose you start Curt Barnes and Jack Davis on it in the morning."

CHECK WEAK SPOTS — Twenty-four hour a day production, seven days a week, makes "preventive maintenance" of heating, piping and air conditioning services essential. Mr. Kirkwood (of Burns & McDonnell Engineering Co., and a member of HPAC's board of consulting and contributing editors) tells here how the Chief and Gard looked over their piping layout and decided to fix up some of the things that might cause trouble and interrupt production. . . . A similar job should be done in every plant where piping is a vital factor

IN CONCLUSION, men, let me repeat that we must maintain a high degree of freedom from shutdowns. Full time production is vital to our defense work. To produce for civilian consumption we must use our remaining equipment to the greatest possible extent. See to it that all equipment and facilities are carefully operated and carefully maintained at all times. Thank you. Good night."

* * *

"Say, Chief, the Old Man sure unloaded tonight. I'll bet there wasn't a foreman present who didn't squirm at least once."

"Right, Gard. There will be a big draft around here if things go wrong, but let's not get caught in it. Let's go over the whole piping layout and remove the weak spots."

"Sounds good on paper, Chief, but how are we going to do it with 24-hour production and some departments on a seven-day schedule?"

"Pick your weak spots first. Fix up those that you can without having to interrupt service. See what you need for the others, and we'll arrange for the time to do the work. Let's have a list with your suggestions. Suppose you start Curt Barnes and Jack Davis on it in the morning."

* * *

Valves

CASE 1—Seats and discs on gate valves cutting out in few months on steam supply lines to pressure cookers.

These valves are used for throttling, a service for which they are not best suited. However, the real trouble lies behind this apparent answer. The amount of steam required during various stages of processing varies considerably. Also, when first installed, only 25 psi low pressure steam was available. When a new batch is started in the cooker these valves are opened wide for 10 to 15 min and even then the 25 psi system has difficulty in getting adequate steam to the cookers. Valves are then well throttled to hold prescribed conditions, but as further ingredients are added, they must be partially opened for a short time to compensate for the incoming cold materials. Then they are almost closed for the balance of the cooking period.

Obviously, gate valves were selected originally because of a combination of low available steam pressure, low loss through the valve,

and lower cost of valve and connections than a globe valve to operate with the same low pressure loss.

The trouble can be eliminated in several ways. First, there is available now a 50 psi steam header that can be enlarged to accommodate the load of the cookers. Globe valves can be installed to supply each cooker.

The second method is based on the fact that a large volume of 20 to 25 psi steam without regulation is required in one part of the processing while only medium or small amounts at lower pressure are required the balance of the time. The answer, in this case, is to bypass the gate valves with a proper sized globe valve. Both valves will be open during the warm-up period. When the materials are at proper heat the globe valve alone will be capable of maintaining adequate flow while throttled and, when practically open, it can handle the added load of additional material added to the batch in the cooker.

Under present conditions, the smaller valve should be more readily obtainable without excessive delay.

Gaskets

CASE 2—Leaking gaskets are showing up regularly on steam lines to engines on grinding mills.

The cause is apparently traceable to vibration, since the gasket leaks are found after periods of high load on the engines. It is during these periods that the lines vibrate materially. It is apparent that the lines cannot handle adequately the flow of steam required under maximum load conditions; the engines are "starved."

For the bulk of the operating time the lines are satisfactory. Therefore, it is not particularly desirable or necessary to replace them with larger lines if other remedies are available. The vibration due to starving is interwoven with the inherent pulsating demand for steam and the required acceleration and deceleration of the velocity of the steam. Inability to produce the necessary acceleration with maintenance of virtually constant pressure leads to even greater steam demands, thus aggravating the situation. The regularity with which these transient conditions repeat is reflected in the more or less flexible piping system. Even though braces and snubbers might reduce the piping vibration, it is not possible to eliminate the main vibration troubles.

The remedy, other than complete replacement by a larger line, lies in providing greater "resiliency" in the steam system. Supply end improvements are eliminated because higher pressure is not desirable. Therefore, the improvement, if any, must be made at the receiving end—a receiver should answer this particular problem.

Fundamentally, the receiver or minor accumulator is a simple contrivance. If selected or sized with care and properly located it will often solve this type of problem. For this type of service the receiver is not expected to store a large amount of steam. It is not supposed to carry the engine load for an appreciable time. On the other hand, it must be capable of caring for the difference between inflow at virtually constant rate and outflow to

Are Radiators Necessary in Air Conditioned Buildings?

THE EDITOR—

In connection with all-year-round air conditioning for offices we sometimes find that bidders will guarantee winter temperature conditions BUT NOT COMFORT unless direct radiators are placed (or left) under windows; the argument being that due to downdrafts along the windows there will be a cold zone which cannot be corrected by hot air from the duct system.

Why is there such a difference of opinion about this point and which group of adherents has the greater weight of experience behind it? We have been trying to get away from our old radiators and now a group of "experts" tells us, in effect, that it can't be done.—L. H., Engineering Dept.

It is difficult to cite any exact data concerning the conditions which make for human comfort because of wide variations in individual preferences. It is easier to cite experience concerning complaints from various rooms in buildings.

In a trust and savings bank, for example, with sealed plate glass windows on three sides, the downdrafts in cold weather across the desks along the outside walls were insupportable, since the air for ventilation entered through ceiling diffusers of a type which tended to increase rather than to diminish the rapid movement of air down along the glass. The matter was corrected by reducing the temperature of the entering air and by adjusting the automatic temperature control so as to increase the temperature of the water in the radiators, thus insuring they were always warm in chilly weather. Had no radiators been installed under the original design, the correction would have been very costly.

In a number of large general offices, the original installation was of the all-indirect type and continual complaint was made of cool air movement downward along the outer walls and at the double windows. The correction involved installing radiators which, by the thermally induced updrafts above them, neutralized the drafts.

It is always less expensive to protect building equipment from freezing by local thermally circulating heat transmitters such as radiators or convectors, than by continuous operation of fans, as would be necessary in all-indirect systems.

I have had wide experience with various types of indirect heating, both by warm air and by steam convectors. Such systems give acceptable results in small homes and in large industrial buildings, but I would install local heat under the windows and along outside walls in my own home, in my children's school room, and in my own office, even though in some rooms the local heat may be furnished by warm air from indirect convectors.—SAMUEL R. LEWIS, consulting engineer, and member of HPAC's board of consulting and contributing editors.

[*Note—Comment on this question from HPAC's readers will be welcomed by the editors.*]

meet engine demands varying widely during each cycle. The receiver should have a volume several times that of the engine cylinder, say, four to six. The installation should be made as close as possible to the engine and the short interconnecting lines should be without unnecessary fittings and of relatively large diameter.

Threaded Joints

CASE 3—Leaking of threaded joints in high temperature steam lines persists at times even when attempts have been made to seal weld. These joints on small lines to power plant auxiliary equipment are difficult to keep tight, especially where pipe wall temperatures are subject to even infrequent heating and cooling.

These joints depend for tightness upon metal-to-metal contacts along the spiral of the threads. If the threads are not true or have blemishes, these joints are likely to leak. When severe temperature changes take place the joint is subjected to considerable working which may open up the normally satisfactory metal-to-metal contact.

Frequently leaking joints of the threaded type are seal welded in the hope that the leaks will be stopped once and for all. Sometimes satisfaction is achieved, but in many cases the leak persists. Welding sets up localized high temperature areas that must cool as the welding is finished. During the welding and cooling process it is entirely possible for differential expansion to be set up with resultant cracking of the weld metal.

The answer to this problem is the use of suitable equipment and technique. There are several lines of welding valves and fittings for small high pressure, high temperature piping; these are often referred to as socket weld valves and fittings. The plain end pipe is inserted into the socket but not to a distance where the pipe is resting firmly on the bottom of the socket. When welded with a small clearance there is practically no chance for differential expansion. Consequently, there is but little chance of cracking the weld metal. In the hands of a competent welder, this joint is an answer to the problem of leaky threaded joints in small lines.

High Temperature Piping

INASMUCH as steam temperatures in modern central stations are approaching those used for the commercial production of hydrogen by reaction between steam and iron, an investigation was undertaken at Purdue University of the corrosion by steam of the various steels which are available for high temperature service. Apparatus was constructed and technique-developed for measuring the amount of corrosion on unstressed specimens due to temperatures up to approximately 1400 F, and pressures up to at least 1600 psi gage. In a paper by H. L. Solberg, G. A. Hawkins and A. A. Potter given at the ASME meeting in Louisville, October 12-15, data were presented showing the effect of various types of surface finish and methods of scale removal, time of exposure to steam at 1100 F for various intervals of time up to 2000 hr, exposure for 500 hr at steady steam temperatures from 1000 F to approximately 1400 F, and temperature fluctuations on the corrosion resistance and spalling of scale on round bars, as well as convex, concave and flat surfaces. Data on the corrosion of cast steels and some special alloys were also given.

The resistance of alloy steels to high temperature steam is greatly influenced by the amount of chromium present. Alloy steels containing 7 per cent or more of chromium are very resistant to corrosion produced by steam at temperatures up to at least 1400 F. The 18-8 stainless steels showed practically no corrosion when subjected to steam up to 1400 F.

Corrosion

CASE 4—Corrosion in hydrochloric acid lines. These lines were designed originally for handling rather weak concentrations. Due to changes in process the concentration now runs quite high. The entire system was made of monel metal and good service was had until the recent change in process.

A discussion with the process engineers develops the definite fact that the higher concentrations are necessary to proper manufacture. Monel metal, according to good authorities, is satisfactory where hydrochloric acid concentrations are about 20 per cent or less. The proper solution of this problem is complete rebuilding of the acid lines using pure nickel pipe, valves and fittings. Silicon bronze might be satisfactory as an alternate material but only where some corrosion can be permitted. In this case added wall thickness is desirable to offset the effects of corrosion.

As in all cases where strong chemicals or chemical solutions are handled, considerable thought must be given both to original materials and to changes of materials should chemical solutions be altered. Operating and maintenance men, metallurgists, engineers and chemists of the operating company usually are well versed in these problems and their proper solution. However, many headaches and much money can be saved frequently by calling in outside consultants and manufacturers' representatives.

* * *

"Well, Gard, these comments and suggestions certainly show up some of our weaknesses."

"Right, and those are not the only ones. I'll have more of them as we go over the plant, but solutions and the write-ups don't come as easily as finding the troubles."

"There is one thing that will be easy this time, though. You won't have to worry about getting the Old Man's approval for most of the expenditures. Forget the change in acid lines until I have taken it up with the bosses. As for the other changes, here is the Old Man's memo covering authorization for materials and labor for all necessary preventive maintenance. I'll O.K. your requisitions for each job. Get 'em done quickly and correctly."

CAN CEILING OUTLETS KAYO Ceiling Smudge?

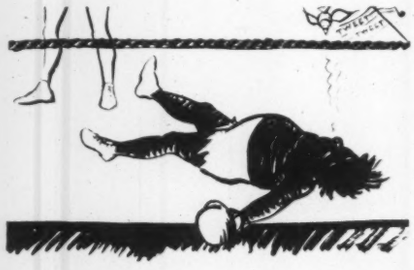
ONE of the greatest recent advances in air distribution came with the discovery of the ceiling diffuser. With this device the engineer is able to insure draftless distribution in many spaces which present difficult problems; the low ceiling basement, the narrow corridor, the brilliantly lighted room requiring great quantities of air—all have been handled successfully with ceiling diffusion.

The use of ceiling air diffusion has sometimes been limited by its unfortunate habit of soiling the ceiling. The dirt deposit is heavier and accumulates more rapidly on rough ceilings of the very type the architect so often selects for acoustical efficiency and artistry. A prominent engineer told us recently that "anyone who uses any type of ceiling outlet on any ceiling that is not smooth and washable is, in my opinion, a fool."

Unsponsored Study

Last fall we started an unsponsored "research project" — not in any laboratory, but out in the field —to discover the cause of smudging and (if possible) its cure. We had no theories. We knew that no one type of diffuser had been 100 per cent successful; but we had received reports that certain jobs were clean. Finally, we knew that certain architects and owners were jumping from one diffuser to another in a vain effort to eliminate smudge.

With this meager start, we investigated job after job. We played no favorites. Every type of ceiling diffuser we could find installed was scrutinized carefully. Our only restriction was that we did not consider the evidence conclusive until a job had been in and running at least eight months. Believe it or not, we did find clean jobs. Wher-



ever we did, we made a thorough investigation, our object being to find how those jobs differed from the dirty ones.

One building was particularly instructive. In that job over 200 diffusers had been installed, most of them in perforated acoustical ceilings. Air was supplied by 16 different fans. Of these 16 systems, all but two were clean. One of these two had extremely dirty ceilings and the other was moderately dirty. Our problem: Why in the same building, even on the same floor, should diffusers deposit dirt at one location and not at another? Obviously, the secondary or room air was not the guilty party since room air at one point was no dirtier than at another point just a few feet away. Why were some ceilings clean and others dirty? Obviously it must be a difference in the quality of the primary air stream. At outlets where the ceiling was clean, the air stream must be comparatively free from dirt. An investigation of filters showed them to be of an unusually efficient type. An investigation of the dirty systems disclosed the simple fact that they had been run without filters for a short period. (Incidentally, in most large cities even a day's run without filters is enough to deposit considerable smudge.)

Culprit—Supply Air

That job gave us a new hypothesis upon which to work, namely, that it is supply air and not room air which is causing most of the trouble. If good filters, properly placed, are used, dirt deposit will not be noticeable within the normal period before redecoration. We continued our investigation and are still doing considerable work along these lines.

To make a long story short, after investigation of many installations, we found that clean jobs invariably had:

1) Good filters.

2) Filters placed on the intake side of the blower or one bank on each side.

3) A tight job with no air bypassing filters at any time.

We also investigated many dirty jobs. In no case did we find all

F. W. Wills and G. F. Nightingale Investigate Jobs Using Ceiling Diffusers, Report on Cause of Ceiling Smudge

CEILING SMUDGE is too often a problem in air conditioned buildings, particularly where the ceiling surface is rough. To find the cause—and also, if possible, the cure — the authors inaugurated an unsponsored "research project," as reported here. They put the blame on the supply air rather than the room air and found that invariably the clean jobs had good filters, filters placed on the intake side of the blower or one bank on each side, and were of tight construction with no air bypassing the filters at any time. For clean ceilings, they recommend an efficient filter for the particular use it is to be put, filters on the intake side of the blower, a tight sheet metal job, no air to bypass the filters at any time, and an outside air intake in just as clean a spot as possible

On a job of this kind, clean supply air is of particular importance

three of these conditions prevailing. In most we found ineffective filters on the duct side of the fans. One job, with two banks in that position, showed dirt spots on the far side of the second bank after only one week of operation. In another job, we found that air was allowed to bypass filters during certain hours of each day.

A careful survey of the evidence accumulated leaves only one conclusion. The supply air, that which passes through the duct, is mainly guilty.

What to Specify

The architect or engineer may have relatively clean ceilings if he will specify:

1) An efficient type of filter for the particular use to which it is to be put.

2) That the filters or at least one bank of filters be placed on the intake side of the blower.

3) A tight job of sheet metal.

4) That no air be allowed to bypass filters at any time.

5) That the outside air intake be placed in as clean a spot and as far above the street level as possible. (This will reduce the load on the filters considerably.)

It will undoubtedly occur to the architect and engineer that it would be far better if the manufacturer would develop a type of diffuser which would eliminate dirt on the ceiling even though these precautions were not taken. We will grant that this would be the ideal solution, for good filters are expensive and so are good tight sheet metal jobs. That solution may be possible at some time in the future but it is a real problem and so far as we know it has not yet been solved.

The obvious solution is to direct the air down from the ceiling just enough to clear it and still not enough to cause drafts. That would be fine but for the fact that air will not always follow our trained arrows.

An important reason for the success of ceiling diffusers in many cases is (according to our experience) the tendency of air in motion to follow a flat surface whenever it is introduced at or near that surface and whenever its direction is parallel to or at an angle of less than 45 deg from that surface. The air delivered hugs the ceiling in a comparatively thin stream, follows down the side walls, across the floors and then back to the center of the outlet in a continuous cycle.

It is this pattern of air motion that makes it so difficult to prevent ceiling smudge and at the same time avoid drafts. It is possible to design a ceiling diffuser which will blow all the air downwards and thus discharge air into a given space without sweeping the ceiling but if this is done, it is almost an impossibility, except in rooms with very high ceilings, to avoid drafts.

Whether an outlet can ever be built which will deliver dirty air to a room and not smudge the ceiling is a moot question. In the meantime, comparatively clean ceilings may be had if these precautions are taken.

Chlorinated Solvents Under Rigid Control

Acting upon advices from the department of agriculture to the effect that the nation's food supply was facing a serious threat as a result of shortages of certain chemicals used by farmers and food warehouses for fumigation purposes, Donald M. Nelson, director of priorities, last month placed under rigid control all stocks of chlorinated solvents. These are defined in the order as carbon tetrachloride, trichlorethylene, perchlorethylene and ethylene dichloride.

A preference rating order assigns a priority rating of A-10 to all defense orders for these solvents, which have not been granted the assistance of a higher rating; and sets up a "ladder of uses" with respect to the supply of these chemicals after defense requirements have been met. Uses for which the rating of B-2 is assigned include charging of fire extinguishers; grain fumigation; the manufacture of refrigerants; the processing and manufacturing of food, chemicals, rubber and petroleum, where substitution of other materials is impractical, and for certain other listed uses. The preference rating of B-8 is assigned to a group of other civilian uses.

PIPING AND CONTROL
for Storage Air Conditioning

William Goodman Gives Recommendations for Jobs Where Compressor Capacity Is Smaller Than Load

STORAGE — There are many places where cooling is required for only a few hours each day. In such cases, it is sometimes possible to make worthwhile savings in air conditioning operating costs by using a chilled water or ice storage tank. This is particularly true if the electrical demand charge is a substantial portion of the operating cost. . . . William Goodman, consulting engineer, The Trane Co., and a member of HPAC's board of consulting and contributing editors, presented last month data to simplify computation of the compressor size and the storage tank required for chilled water systems. This month he gives information on the piping, tanks, and control, and shows derivations of the various equations which he has developed

THE diagrams on pp. 561, 562 and 563 of the September HPAC showed a single circulating pump for the storage refrigeration systems described. A single pump is perfectly satisfactory for installations in which the storage tank is used only to prevent short cycling of the compressor; that is, installations in which the running period of the compressor is never greater than the actual cooling period. However, for installations where the running time of the compressor is greater than the actual cooling period, it is more desirable to use two separate pumps with the storage tank, as illustrated here in Fig. 1. A pump must be used to circulate water whenever the compressor is running. If a single pump is used, it must circulate the water against the entire head of the system during the water cooling period when no conditioning is required. On the other hand, if a separate pump is used for circulating the water between the water chiller and the tank, as shown here, more economical pump operation will be obtained.

A further disadvantage of circulating systems like the ones illustrated on pp. 561 and 563 in September is that the storage tank must be capable of withstanding the entire pressure of the system. Because of the large tanks that generally are necessary in systems where small compressors are used, tanks open to the atmosphere are usually more economical to install. When an open storage tank is used, the coils can be installed in the storage tank or, as is more commonly the case, a separate chiller can be employed. In either case, more economical operation will generally be obtained if a separate pump is provided to circulate the water between the storage tank and the water chiller as in this month's Fig. 1.

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Water Control

A three-way valve is used for each air conditioning unit, or for an entire group of units in a single zone. The thermostat actuating the three-way valve for the air conditioning unit should be situated in the return air stream. The amount of water supplied to the conditioning units will then be governed by the dry bulb temperature in the conditioned spaces.

In addition, a three-way valve should always be used to bypass water around the storage tank and

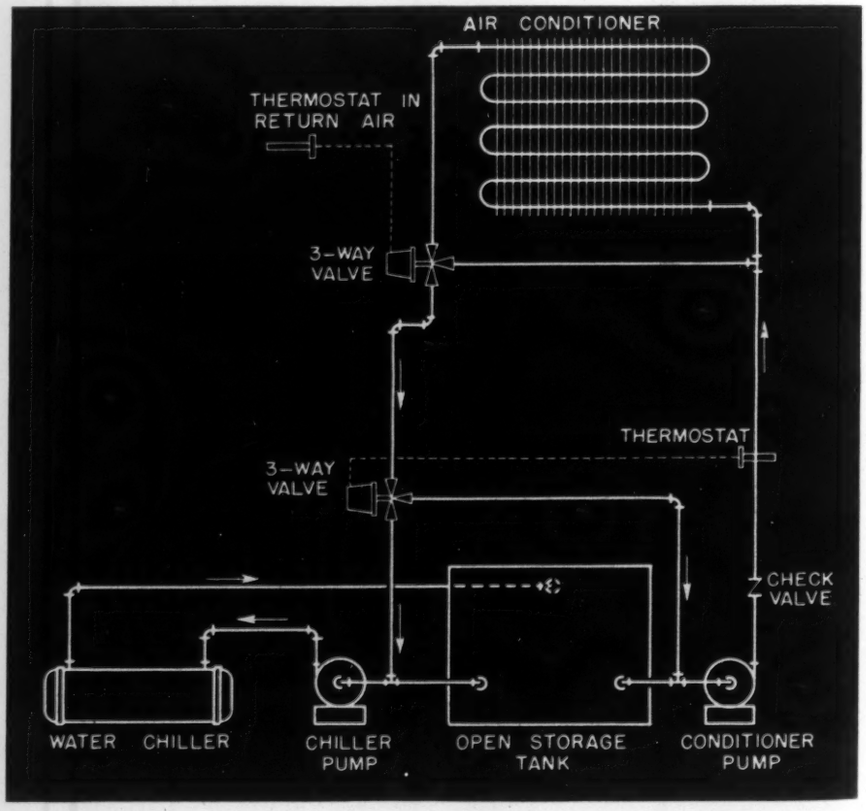


Fig. 1—Arrangement of piping and controls in storage tank system

into the suction of the conditioner pump. This valve should be controlled by a thermostat in the water discharge line in the conditioner pump. By means of this thermostat and three-way valve, water will be supplied to the air conditioning units at a constant temperature regardless of the temperature of the water in the storage tank. This is highly desirable, not only because better temperature control will be obtained if water at a uniform temperature is supplied to the air conditioning units but also because greater economy will be achieved. At the beginning of the conditioning period, if water at the low temperature in the storage tank is supplied to the air conditioning units, excessive dehumidification will occur with resulting unnecessary increase in compressor operation.

In Fig. 1, water leaving the three-way valve at the storage tank is discharged either into the suction of the conditioner pump or the suction of the chiller pump; the water is not discharged back to the storage tank directly. By connecting one of the lines from the three-way valve at the storage tank into the suction of the chiller pump, water at a higher temperature will be supplied to the water chiller. In this way, during periods when the three-way valve is sending water to the suction of the chiller pump, the compressor will operate at a higher suction temperature and with resulting greater capacity and economy. If desired, the three-way valve can be arranged to close against the suction line to the chiller pump whenever the conditioner pump stops. This will eliminate any possibility of creating a slight circulation of chilled water through the coils of the conditioning unit when the fan is off.

Tank Baffles

A side view of the tank illustrated in Fig. 1 is shown in Fig. 2. Baffles are frequently used in these storage tanks in order to assure thorough mixing of the water. W. F. Friend states [see Refrigerating Engineering, February, March and April, 1938] that "Preferable arrangement of baffles is vertical, placing them across width of tank at 5 to 6 ft intervals. Alternate baffles then project above normal static water level and terminate about 1.5 ft above tank bottom; intermediate baffles start 1 ft below water level and extend to floor. Provision should be made for venting the air spaces between alternate baffles above water line, and small openings are needed at bottom to permit flushing out sediment toward sump at one corner of tank floor." Thorough mixing of the water must take place in the tank or the storage effect may be greatly decreased.

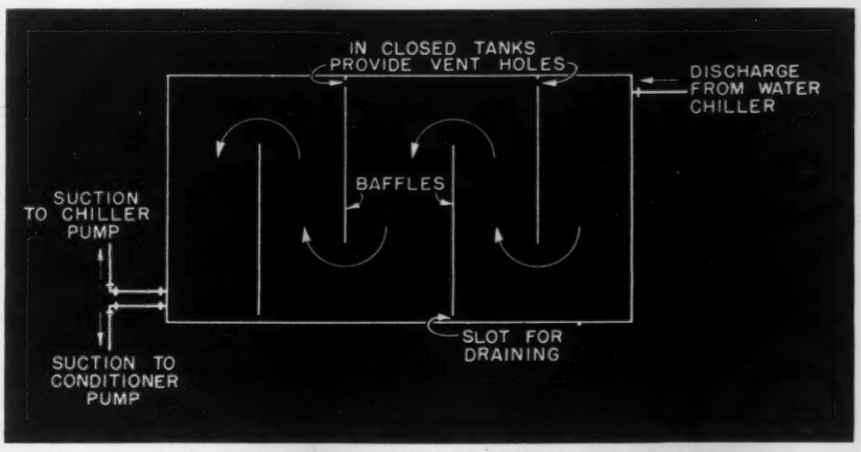


Fig. 2—Side view of storage tank showing baffle arrangement

For installations of the type illustrated on p. 562 in the September HPAC, in which the coil is submerged in the tank, a single pump may prove satisfactory if the head against which the pump must work is not too great. However, if the external head is of any magnitude at all, it will frequently prove more economical to provide a separate small pump simply for circulating water throughout the tank alone. Instead of providing a separate pump, a bypass valve can be used to bypass the water around the air conditioning equipment whenever cooling is not required in the conditioned spaces. However, in this case, the pump efficiency may be poor. It is necessary to check the efficiency of the pump when working against both heads to determine whether a single pump will be satisfactory. Whether an extra pump is warranted for an installation like this must be decided on the merits of the individual case.

Incidentally, for a storage system with submerged coils, there is—strictly speaking—no need to operate a pump at all during the period when no cooling is required in the conditioned spaces. If sufficient cooling surface is provided, the water in the tank can be cooled by the submerged coil without any forced circulation. However, forced circulation of the water in the tank is highly desirable not only because of the improved heat transfer from the coils to the water, but also because the water in the storage tank can be maintained at a more nearly uniform temperature.

Compressor Control

The compressor should be started automatically whenever cooling is provided in the conditioned spaces. The compressor can be started automatically either by the starting of the fan circulating chilled air through the conditioned spaces, or by the starting of the conditioner pump which circulates chilled water between the storage tank and the air conditioning unit. The compressor can then be allowed to run continuously as long as air conditioning is being provided.

A thermostat should be installed in the storage tank. This thermostat should stop the compressor when the water in the tank has finally been cooled to the predetermined low point, say, approximately 36 to 40 F. In this way, whenever conditioning is required, the compressor will be sure to run. However, even after the air conditioning unit has been stopped, the compressor will continue to run until the water in the tank has been cooled to a sufficiently low temperature.

It is important to remember that the compressor must be started whenever the air conditioning unit is started. If the compressor is not started, the conditioning load will

heat the water in the storage tank to too high a temperature before the need for conditioning has ceased. The compressor on starting would then be faced with the double duty of simultaneously cooling the water and the conditioned space which would, of course, be impossible. For this reason, the compressor should be started whenever the air conditioning is required and should be allowed to run until the thermostat stops it.

Control of the compressor as described above can be accomplished by wiring the tank thermostat in parallel with the holding coil of the starter of either the circulating fan or the conditioner pump. In this way, either the starting of the fan or pump, or a rise in water temperature, will start the compressor, and the thermostat must always be satisfied before the compressor can stop. The thermostat should have a small differential, say 2 deg or less, so that if conditioning is not required for some time during a cool period, a tank of cold water will always be available for a sudden hot spell.

For the rare occasion when the air conditioning load is less than the compressor capacity, the water would cool toward the freezing point, and under extreme conditions, might cool to the freezing point. To prevent this, an auxiliary low limit thermostat should be inserted in the pipe line leaving the water chiller. Whenever the water leaving the chiller falls to about 35 F, the compressor should be stopped.

In order to be sure that water is circulating through the chiller whenever the compressor is running, it is desirable to insert an orifice or pitot tube in the chiller pump discharge line. A differential switch should be connected across the orifice or pitot tube as mentioned in my September article. When water is flowing through the chiller, the pressure difference across the orifice or pitot tube can be used to actuate the differential pressure switch to start the compressor. If a differential pressure switch is used, starting the fan or closing of the tank thermostat would not start the compressor directly. Instead the pump circulating water through the chiller would be started first and the differential pressure switch would then start the compressor.

The orifice and differential pressure switch offers protection not only against complete stoppage of water flow, but also against a decrease in flow to a dangerously low value. If the flow should fall to a dangerously low value due to an obstruction, the falling pressure difference across the orifice would trip the differential pressure switch and stop the compressor. Incidentally, in each pump suction line strainers should be installed to remove dirt from the water.

Motor Overloading

To cool the warm storage water when the system is first placed in operation each season, the air conditioning equipment need only be started and run for a few minutes. If the control system described in the previous section is used, starting the conditioning unit will start the compressor and chiller pump.

There sometimes appears to be some misapprehension in regard to the overloading of the compressor motor when the system is first placed in operation. As shown in the articles on horsepower requirements of refrigerating compressors which ran in HPAC from December, 1940, to February, 1941, inclusive, the horsepower required by any compressor is a maximum at

"QUOTED"

SMOKE

AMONG the chief causes of failure of initial smoke campaigns to develop into permanent smoke abatement programs have been the following: (1) Misconceived ideas of the real problems involved. (2) Misconception of the time element required for permanent smoke abatement. (3) Lack of sufficient properly informed public sentiment. (4) Lack of active and forceful individuals and organizations backing the program. (5) Lack of competent men in the municipal smoke abatement department. (6) Improper organization of the smoke department. (7) Disregard for the economic situation of the community. (8) Ineffective smoke prevention educational program. (9) Politics. (10) Lack of sound political judgment on the part of those active in smoke abatement. (11) Lack of experience in municipal smoke abatement. (12) Improperly drafted ordinance. (13) Municipally owned buildings failing to comply with the ordinance. (14) Lack of proper financing of the program. —*T. A. Marsh, engineer, Iron Fireman Mfg. Co., at the fall meeting of the ASME in Louisville.*

BOUILLON

ONE of the major problems confronting the manufacturer of bouillon cubes — preventing the cubes from sticking to the machines which compress and automatically wrap them — has been solved through air conditioning.

Close control of humidity and temperature in the beef bouillon cube manufacturing plant of the Romanoff Caviar Co. in New York City has speeded up operation considerably and aided materially in turning out a better product.

The Romanoff company is one of the leading firms dealing in imported caviar and similar food specialties. Beef bouillon cubes, in recent years, had grown to become one of their biggest items and increased production presented quite a problem.

Material used in this food product is extremely hygroscopic. While a direct means of moisture or humidity removal and control was therefore indicated, it was necessary to conduct tests to determine the extent of humidity control to produce optimum conditions.

As a result of these tests, a rela

Symbols

H_C = refrigerating capacity of the compressor, tons
H_L = cooling load, tons
H_S = thermal energy stored in tank, Btu.
H'_S = thermal energy stored in tank, ton-hours.
θ_C = running time of compressor, hr.
θ_L = duration of cooling load in conditioned spaces, hr.

some one suction temperature. *Above* and *below* this suction temperature, the horsepower required decreases. For chilled water storage systems, the compressor would ordinarily be selected for approximately a 30 F suction temperature. For this suction temperature, the "Freon" compressor requires almost its maximum horsepower. So close are the power requirements to the maximum horsepower that any actual motor selected for the compressor would probably have more than sufficient capacity to meet the maximum power requirements of the compressor. However, the important point to remember here is that for operation at suction temperatures above approximately 30 F, the horsepower drawn by the compressor would probably be considerably less than the horsepower at approximately 30 F. Therefore, no extra devices are required to prevent overloading of the compressor when it is used to cool a tank of warm water at the beginning of the cooling season.

Derivations

When the compressor capacity is less than the cooling load, the running time of the compressor must be greater than the duration of the cooling load. The refrigeration produced during the period when there is no cooling load is stored. The period during which there is no cooling load is $(\theta_C - \theta_L)$. Hence,

$$H'_S = H_C\ (\theta_C - \theta_L) \quad \ldots\ldots\ldots [a]$$

Ultimately, all of the cooling load must be taken care of by the compressor. Consequently,

$$H_C\theta_C = H_L\theta_L \quad \ldots\ldots\ldots\ldots\ldots [b]$$

This is the same as the first equation on p. 631 of the October HPAC.

Eliminating H_C between equations (a) and (b),

$$\frac{H'_S}{H_L} = \theta_L\left[1 - \frac{\theta_L}{\theta_C}\right] \quad \ldots\ldots\ldots [c]$$

For a cooling load of one ton, $H_L = 12{,}000$ Btu per hr. Also $H_S = 12{,}000\ H'_S$. Hence, the storage required per ton of cooling load is

$$H_S = 12{,}000\ \theta_L\left[1 - \frac{\theta_L}{\theta_C}\right] \quad \ldots [d]$$

This is the second equation given on p. 631 in October.

To obtain the equation given on p. 633 of the October HPAC, write equation (a) above as

$$H'_{S_1} = H_C\ (\theta_{C_1} - \theta_L) \quad \ldots\ldots\ldots [e]$$

where H'_{S_1} is the storage capacity required for a compressor running time of θ_{C_1}.

Now for the same compressor capacity and cooling load period, a storage capacity, H'_{S_2}, will be required for a compressor running time of θ_{C_2}, or

$$H'_{S_2} = H_C\ (\theta_{C_2} - \theta_L) \quad \ldots\ldots\ldots [f]$$

Dividing (f) by (e), and using the relationship $H'_{S_2}/H'_{S_1} = H_{S_2}/H_{S_1}$,

$$\frac{H_{S_2}}{H_{S_1}} = \frac{\theta_{C_2} - \theta_L}{\theta_{C_1} - \theta_L} \quad \ldots\ldots\ldots\ldots [g]$$

This is the equation given in the text.

tive humidity of 30 per cent at a temperature of 83 deg was selected as being ideal. To maintain this condition, a silica gel dehydrator was installed for direct control of humidity, and a mechanical refrigeration compressor provides direct and independent control of the temperature.

This is believed to be the first installation of its kind in the bouillon cube industry. The significant feature of the installation is that separate control of humidity is accomplished through use of the separate dehydration or dehumidifying equipment.—*Edward T. Murphy, vice-president, Carrier Corp.*

BROKEN

ALL previous Chicago records for the sale of central plant air conditioning systems and electric room coolers were broken in the first nine months of this year. A total of 406 central plant installations with an aggregate capacity of 12,267 hp were contracted for. This compares with 347 systems with a combined rating of 7571 hp sold in the corresponding period of last year and 355 totaling 11,381 hp in the former record nine months of 1937.

A total of 1582 electric room coolers were sold in the first nine months of the year to chalk up an increase of 61 per cent over the previous record in 1940.

September this year was the highest in history in sales of both central systems and room coolers.—*Commonwealth Edison Co.*

NOSE KNOWS

THE eye and the nose of the safety engineer constitute reasonably accurate measuring devices for the detection of hazardous dusts, gases and fumes in industrial plants. Although there are delicate and precise measuring instruments available for detecting the presence of dangerous dusts and vapors, it is not always necessary to know the presence of dust particles in terms of "parts per million" to know if the air is unhealthy, or at least unpleasant for workmen.

If the safety engineer can see dust floating in the air near the workman he can be pretty sure it is none too healthy or pleasant for the workers. There are many more dust particles than can be seen and the workman is breathing them during his entire working day.

The same is true of invisible fumes and vapors. If they cannot be seen they may be detected by their odor. If a strong odor of the toxic acids can be detected, it is well to make provisions for dissipating the gas. I don't believe it is good industrial policy to ask a workman to stay all day in such unpleasant odors even though there is no particular health damage provable.—*S. W. Gurney, laboratory director for the Liberty Mutual Insurance Co., at the National Safety Council, October 9.*

Giant Plants Build GIANT PLANES

Heating and Ventilating the New Boeing Plants for Manufacture and Assembly of Flying Fortresses

WITH the 41 acre bomber plant of the Boeing Aircraft Co. completed and in production at Seattle after a record 12 month expansion program, the Austin Co., engineers and builders, have started construction of a 1,320,600 sq ft addition to Boeing's new bomber plant at Wichita, Kansas, which will soon match in size the enormous Seattle development. These twin plants, specially designed for speedy, large scale production, are being used for the manufacture and assembly of Boeing B-17E Flying Fortresses.

Approximately 42 miles of piping carries steam, air and water direct to every position in the Seattle plant where such facilities are needed.

Forty-six miles of pipe required by the sprinkler system have 20,000 individual sprinkler heads and 75 tons of fittings, exclusive of gate valves which weigh seven tons. Water for this system is furnished in part from a 22 in. city main and the remainder from a 150,000 gal tank with a motor driven fire pump of 1500 gpm capacity.

The main utility lines are carried throughout the plant through a series of tunnels, auxiliary trenches and underground passageways. These 10, 13 and 15 ft wide passages, besides being used for utility lines above the headroom plane, are the means by which the shop employees have access to all underground washrooms, toilets, transformer rooms, and to any department in the plant in which they may be employed without passing through or in any way interfering with the operations of any of the other departments.

Executive Office Building—Heating and ventilating is provided by a direct-indirect system. The tempered, washed and filtered supply air has a minimum of one-third outside air, the balance being recirculated. The air is changed throughout once every 10 minutes, except in kitchen and dining rooms where the air is changed every three minutes, and in toilets where it is changed every six minutes.

There are four exhaust systems: (1) Return air system from offices, etc., for recirculation. (2) Toilet exhaust system. (3) Dining room and kitchen exhaust system. (4) The basement kitchen has a fireproof exhaust with an outlet approximately 125 ft from the outside air intake.

The fan room is on the roof and the equipment is under pneumatic control. The air is supplied to the individual rooms through high velocity air diffusers, and the air is returned through these same diffusers to the fan room for recirculation. The direct heating is a vacuum system consisting of convectors supplied with steam through zone controlled valves.

Engineering Building—The heat-

ing and ventilating is a direct-indirect vacuum system. The ventilation is based on 30 cfm per person, allowing a minimum of 10 cfm per person of outside air. There are four exhaust systems: (1) Return air system for recirculating. (2) Toilet exhaust system. (3) Blueprint room exhaust system. (4) Chemical exhaust system which removes the foul air from the laboratories and photographic rooms. The blueprint room has a separate supply system.

The fan room is above the trusses in the adjoining warehouse building. The air is supplied to the various rooms and areas through high velocity air diffusers and returned to the fan room for recirculation through grilles and ducts at the floor lines throughout the building. The direct heating is a vacuum system consisting of convectors, manually controlled individually. The reheat for each floor is thermostatically controlled.

Assembly Sections—These sections have a series of underground trenches, in which at regular intervals, through hinged checkered plate covers, electric power for welding, etc., also compressed air, are available with twist lock connectors. The temperature maintained is 65 F with wall type, floor type and ceiling type unit heaters. The ventilation is controlled through motor operated monitors.

Primaries and Basic Primaries Sections—These sections have power and compressed air available through twist lock connections at practically every column. The temperature is maintained at 65 F.

Warehouses—The north warehouse was designed for a temperature of 55 F, while the south warehouse was designed for 65 F. Ventilation is through steel sash and monitors. The areas are heated with ceiling type projection heaters and wall type units.

Boiler Room—There are five 250 hp boilers, complete with oil burning sets, soot blowers, steam driven feedwater pumps, feedwater heater and hot water supply tanks. There are three 5000 gal fuel oil tanks so arranged that they can be filled direct from tank cars.

Miscellaneous Buildings—Personnel, hospital, gate, time office, paint storage buildings, and acetylene, oxygen, hydrogen generating buildings are all supplied with the various utilities through underground concrete waterproof trenches. Temperatures are maintained at 70 F with high pressure steam convectors.

The dope rooms are designed for a temperature of 95 F. The booths are ventilated by exhausting 20,000 cfm through each booth area. This air is taken from the surrounding floor areas through filters and reheated by unit heaters.

The finishing rooms are similar excepting that working temperature is maintained at a minimum of 75 F.

The north and south distributing aisles are heated with projection type heaters and the temperature is maintained at 55 F.

Anodic Room—The anodizing tanks each have a separate exhaust system complete in itself, exhausting all fumes arising from tanks to the outside. The anodic room has a dryer system capable of drying 6000 sq ft of surface every 20 minutes.

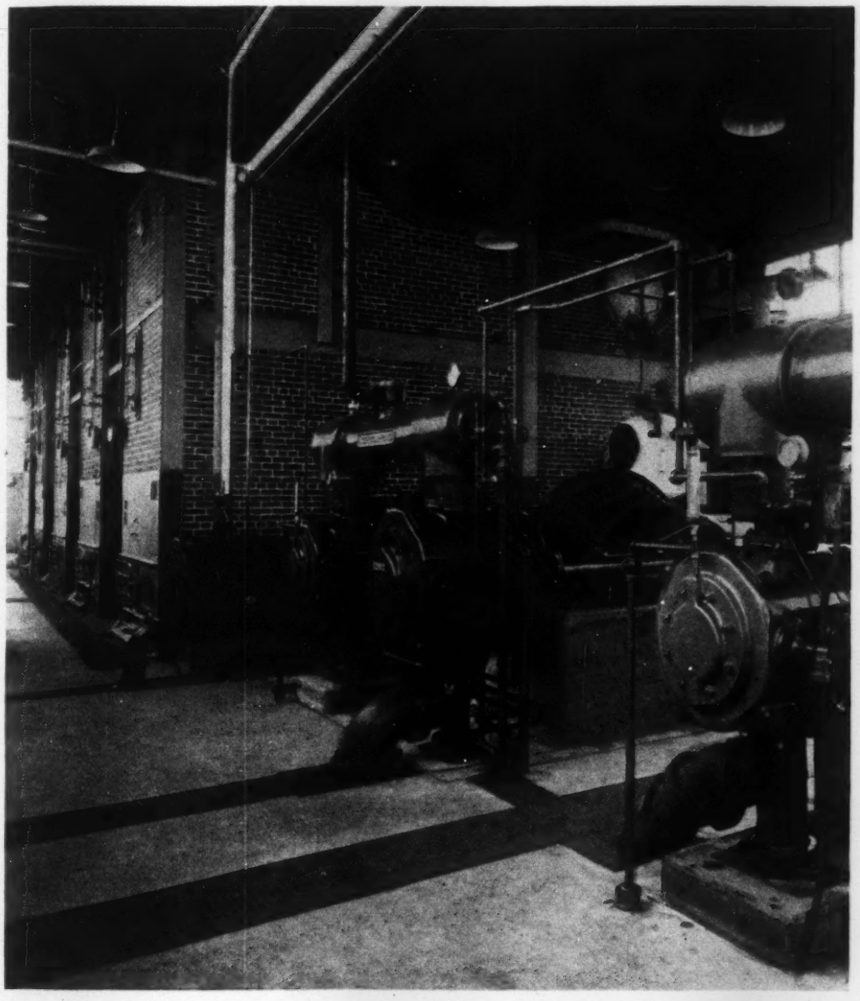
Steel Heat Treat Room—The various furnaces have their hoods connected to a separate exhaust system.

Washrooms are underground areas directly connected with the tunnels with arched openings.

These washrooms are heated and ventilated with outside air through indirect heating surface and through a fan to a duct system which distributes the air evenly throughout the room, the air leaving through the arched openings into the tunnels, from where it is exhausted by being drawn into the various underground toilets and transformer rooms which have individual mechanical exhaust.

There are four of these washrooms, each one having a system supplying 8000 cfm of outside air, heated and controlled to maintain a temperature of 70 F in the washrooms and tunnels.

Inside the boiler house, which provides heat and compressed air for Boeing's plant No. 2 at Seattle, there are five 250 hp oil fired boilers and two 900 cfm air compressors. All piping throughout the boiler house and plant is readily accessible through steel floor plates. There are underground access tunnels throughout the entire plant



"Open for Discussion"

Communications to the Editor

WHY PANEL HEATING?

THE EDITOR—

I have read with considerable interest F. W. Hutchinson's article in the October HPAC on this subject of panel or radiant heating.

After many years of experience with radiant heating, I agree with Mr. Hutchinson that with radiant heat a room temperature of 5 to 8 F lower than is normally set for convected heat is found to be perfectly comfortable, provided the radiant panels are sufficient in quantity and correctly located. It may be that some installations can be found that do not substantiate this, but I venture to say that any radiant heat system which will not give equal comfort conditions at a lower air temperature than that necessary by a convected heat system has been provided with insufficient or faulty radiant surfaces which do not take care of the radiation from the human body. This protection against radiation loss is the greatest factor on the side of radiant heating and is the main reason why we can have a lower air temperature and feel perfectly comfortable.

In addition to this, however, there is a distinct advantage because of the higher relative humidity which is a consequence of a lower air temperature. In cases of extreme temperatures, this higher relative humidity may not appear to be very substantial, but even a small percentage of increase at such extreme temperatures will make quite a lot of difference to human comfort.

There is, I think, another important factor which must be taken into consideration and that is the metabolism of the human body. It seems to be well established that as the temperature of the air is lowered, a greater chemical action takes place within the body and more heat is produced. In other words, the fire within burns more brightly and heat is generated more rapidly as the temperature drops to 60 F than is the case where the air temperature is high. According to the best information available it seems to be substantiated that with a high air temperature the organs of the body relax, especially those organs generating heat; consequently, a smaller amount of heat is generated, which explains the reason why some people often feel chilly in a room temperature of 80 F or above. With a properly installed radiant heat system with sufficient radiant surfaces, it is found that the human body can function more correctly and generate sufficient heat to maintain a very comfortable condition with air temperatures as low as 58 to 60 F.

An English medical research committee, after a number of exhaustive tests and investigations, came to the conclusion that the agility of students is much higher and a greater activity of the mind and muscles is shown in a cooler atmosphere with radiant panels to protect the body against too much radiation loss. Therefore, I think we must look to this function of metabolism for one of the reasons why the body can be comfortable with a much lower air temperature. —T. NAPIER ADLAM, Sarco Co.

AIR DISCHARGE FROM NARROW SLOTS

THE EDITOR—

I was gratified to have my article on the discharge of air from narrow slots reviewed by such eminent authorities as Professor Fellows and Professor Nelson on pp. 558-560 of the September HPAC. I noted with interest the formula in Professor Fellows' review for obtaining the static pressure of air at any point in a duct discharging the air through slots. Coming from the late Professor Goodenough, I have no doubt of its accuracy; but like several other formulas I checked, it seems to me to be entirely too intricate for the average ventilation engineer to use.

I acknowledge that the formula for ascertaining friction loss in such a duct as given in my article gives approximate results only, especially if used for various points along the duct instead of at the end. However, it seems to work out pretty well with actual field tests I have made and until something better is offered, it should be useful in many cases.

I was interested in what Professor Fellows said about calculating static pressure regain in a duct of constant cross sectional area and of constantly diminishing air volume. In all the data on static pressure regain I was able to find, it is assumed that the air mass is constant and the duct area changes. It would be interesting to have some experiments made with suitable laboratory facilities in order to get definite information about this subject. It would also be interesting to know the exact effect of turning vanes placed in the slots: How much of the velocity pressure of the air flowing in the duct is available for useful discharge work when vanes are used and how do they affect the discharge static loss? I have had a number of opinions on these subjects, with much variance in the views expressed.

Regarding the formula to calculate the throw of air from slots, to which both discussers took exception, it was given to me by H. M. Hendrickson, of Los Angeles. It is based on the assumption that air discharging from a narrow slot without turning vanes with a velocity of 1200 fpm will have a throw of 20 ft.

The empirical formula is

$$Cfm = \frac{20 \times l \times h}{3} \quad \ldots\ldots\ldots [1]$$

where Cfm = cubic feet per minute of air volume flowing; l = length of slot, in.; h = height of slot, in.

Equation 1 may also be written

$Cfm = 20/3 \times$ area of slot in sq ft $\times 144$

We also may state that

$Cfm =$ Coefficient of discharge $\times$ discharge velocity at slot in fpm $\times$ area of slot in sq ft .....[2]

Two expressions both equal to the same expression are equal to each other. Hence,

Coefficient of discharge $\times 1200 \times$ area in sq ft $= 20/3 \times$ area in sq ft $\times 144$ .....[3]

Solving equation 3 for the coefficient of discharge, and assuming the area to be 1 sq ft, the coefficient is found to be 0.8. Then

$Cfm = 0.8 \times 1200 \times 1$.....[4]

The throw of air is a function of the discharge velocity and the square root of the slot area. Therefore, we can say that

Throw, in ft = a constant $\times$ discharge velocity $\times \sqrt{\text{area}}$.....[5]

In terms of the assumptions for equation 1 this will read

$20 =$ a constant $\times 1200 \times \sqrt{\text{area}}$ .....[6]

For a unit area of 1 sq ft

$20 =$ a constant $\times 1200 \times \sqrt{1}$ .....[7]

and

$$\text{a constant} = \frac{20}{1200 \times \sqrt{1}} = \frac{1}{60} \quad \text{.....[8]}$$

Then

$$\text{Throw} = \frac{1}{60} \times \text{discharge velocity} \times \sqrt{1} \quad \text{.....[9]}$$

$$= \frac{\text{discharge velocity}}{60} \quad \text{.....[10]}$$

From equation 2 we may state that

$$\text{Discharge velocity} = \frac{Cfm}{\text{coefficient of discharge} \times \text{area}} \quad \text{.....[11]}$$

or

$$\text{Discharge velocity} = \frac{Cfm}{0.8 \times 1} \quad \text{..[12]}$$

Substituting this value for the discharge velocity in equation 9 we have

$$\text{Throw} = \frac{\sqrt{1}}{60} \times \frac{Cfm}{0.8 \times 1} = \frac{Cfm}{48} \quad \text{.....[13]}$$

The chief exception that I take to this formula is that the coefficient of discharge 0.8 is high. It would be safer to use, say, 0.65. Working out the formula again from this assumption, the throw of course would be affected. If the same slot area were taken and the fan run at the same speed, then the cfm would be greater on account of the smaller *vena contracta* effect at the discharge. That is if the fan continued to run at the same rpm, and assuming the discharge pressure the same, more air would be discharged from an orifice having a higher coefficient of discharge, and greater cfm would tend to increase the throw. The foregoing is based on no turning vanes being used.

However, the formula seems to give fairly good results in field tests, considering its limitations as mentioned by Professor Nelson. Professor Nelson's statement that the short throw, common to the narrow slot discharge, is due to the large amount of air stream surface, exposed to the resistance of space air, is, of course, correct. I mentioned this in my article. His observation that if the air can be discharged so that it slides along the surface of the ceiling that its throw will be lengthened is also correct. However, in the case I cited, the air was projected so that it bounced from the ceiling and the throw was shortened.

I examined another formula for throw, credited to Professor C. O. Mackey, but this formula, like that of Goodenough for friction loss, was so intricate that I deemed it impracticable for average use. It

CLEVER IDEA—Each of the 40 apartments in the Brooklawn Apartments, Hartford, Conn., is year 'round air conditioned by an individual unit conveniently installed in a closet space. Note the bookcase built into the door, as well as the intake louvers for return air. The units are controlled by a combination heating and cooling thermostat. Hot water is the heating medium, and for cooling the same centrifugal pumps circulate deep well water at a temperature of 53 F. The closets housing the units are generally situated in a central section of each apartment, and are placed one over the other to simplify piping. Two air outlets, directed toward the outside walls, are used in each apartment living room

Photo courtesy Trane Co.

is a rational formula based upon a spread of the air stream through a total angle of 14 deg, constant momentum and a velocity of the air at the end of the throw of 50 fpm. Values of the actual throw will be slightly less than that obtained from this equation, according to Professor Mackey, and a multiplying factor of 0.8 will give a good approximation for the actual throw in most cases. This formula is:

$$\text{Throw in feet} = 8.33\left[0.06\,(a^2+b^2)+0.24\,ab\left(\frac{V^2}{2500}-1\right)\right]^{1/2}-2.05(a+b)$$

where a = shorter dimension of the slot, ft; b = longer dimension of the slot, ft; V = discharge velocity of air at slot, fpm.

The above equation assumes that the discharge air is at the same temperature as space air.

In this connection, I believe a standard definition for throw should be adopted. By some, it is said to be the perpendicular distance the discharge air travels from a slot or outlet until it starts to drop. By others it is said to be the total travel of the discharge air away from the slot, disregarding its rise or fall. This is the logical definition and is the one used in my article.

As stated before, I would like to see some experimental work done on this subject by competent authorities with proper recording equipment and facilities.—F. F. Stevenson.

PRE-EMPLOYMENT EXAMINATIONS

The Editor—

I thought the article entitled Ignorance Is No Excuse, by Lawrence R. Bloomenthal in the September HPAC, to be of especial interest. It seems to be the prevalent theory of some managements that as long as they provide the usual safeguards for the health of their employees, that is the extent of their responsibility. They do not seem to realize that their responsibility does not end at this point but that further education and enlightenment of the workers with respect to the intelligent use and care of these safeguards is also part of that responsibility.

Recently, for example, a judgment was issued against a plant where safeguards were provided, but the employee was not adequately educated to the use of these safeguards. An occupational disease had subsequently resulted. I think that this article, therefore, will be an excellent reference for me in pointing out to some of our managements that their responsibility does not end with just providing health safeguards as far as their legal interest in occupational disabilities is concerned.

A program which the Los Angeles division of industrial hygiene is encouraging is the necessity of pre-employment physical examinations in industries. Many plant managements are not aware that the reactivation of an existing occupational disability by any process or exposure can be the grounds for legal suit against the management, despite the fact that this exposure ordinarily may not be sufficient to cause a similar disability in an ordinarily healthful individual.

A case similar to this occurred recently in one of our aircraft plants in which a tubercular individual supposedly cured of his condition developed a recurrence of his illness due to a dust exposure which ordinarily would not have caused this condition to appear in a normally healthful individual. Although this particular company had already inaugurated a program of pre-employment physical examinations, the misled sympathy of the examining physician allowed this man to work in the dusty atmospheres which caused the reactivation of his tuberculosis.

There are similar situations which I might go on to enumerate, but I believe that this is representative in pointing out what I mean. I would like to have seen a complete discussion by Mr. Bloomenthal regarding the legal responsibilities of plant managements as far as pre-employment physical examinations are concerned, as I believe this to be a problem of wide interest to those of us working in industrial hygiene and to industry in general.—Yale Rosenfeld, engineer, division of industrial hygiene, department of health, City of Los Angeles.

A STEP IN REVERSE

The Editor—

I read with keen interest all the letters in this column, for here is where the average man can make known his likes and dislikes. I noted a letter from J. H. Walker in September in which he asked "Why panel heating?" The same question has been in my mind for some time —*why?*

If all you are after is heat with no humidification or ventilation, then of course a panel installation is fine. However I think panel heating is a definite step in reverse. What we are after is cleaner, more healthful air in our buildings, along with body comfort. It makes no difference whether it be for summer or winter, if we are to get such a condition we cannot use the panel method. Some wise guy is going to say use a split system. What for? Why not do the job correct in the first place?

One other point has been bothering me on this subject. Whenever mention is made of panel heating the article is usually headed "Panel Heating and Cooling," but nothing is ever said about the cooling cycle. How come? I'll tell you why no one mentions it. It is a flat failure. I make this statement from actual experience. You can cool by the panel method but it had better be a meat storage house and not a building where humans are going to live and work.

If we are going ahead, let's forget all about panel heating and cooling and sell our country's buildings year 'round air conditioning using plenty of outside air—filtered, cooled or heated, humidified or dehumidified, to give real healthful comfort. —Will D. Sampson, consulting engineer.

AIR CONDITIONING INDUSTRY OFFERS PRIORITY PROGRAM TO GOVERNMENT

Washington, D. C.,
October 22, 1941

Office of Production Management
New Social Security Building
Washington, D. C.

Gentlemen:

We herewith present a program for a curtailment of between 35 per cent and 50 per cent in the total production of the air conditioning and refrigeration industry, by eliminating non-essential civilian uses.

Already our effective part in the defense program is being seriously hampered by the inability of our industry (industrial and commercial) to obtain necessary materials. We shall briefly explain why and offer constructive suggestions for your consideration.

Refrigeration is the process of lowering the temperature of a substance by mechanical, chemical, or physical means.

Air conditioning is the process of controlling the conditions of air as regards temperature, humidity, circulation, ventilation, and cleanliness. In summer, this includes refrigeration to reduce temperature and to reduce the moisture content of the air. In winter, it includes heating to increase temperature and humidifying to increase the moisture content of the air.

The industrial and commercial air conditioning and refrigeration industry (*excluding domestic refrigerators*) may be grouped into three broad divisions:

a) Refrigerating and air conditioning machinery.

b) Fixtures (cabinets, cases, and other fixtures).

c) Accessories and parts.

The above will hereinafter be referred to as the "AC&R industry."

The undersigned committee has been selected by the OPM as representative of the entire AC&R industry, including the remote branches. This committee includes responsible members of all three divisions; it enjoys the confidence of all branches and has the right to speak on their problems. Evidence of this fact is the vital and not-previously-available information supplied, at the request of this committee, for 163 companies, large and small, representing practically 100 per cent of the AC&R industry (Appendix I).

The size of our industry and the extent to which it affects almost all other industries, as summarized in Appendices I and II, will no doubt surprise those who have not previously had occasion to be informed.

39.9 Per Cent Is Defense

The AC&R industry is now doing an average of 39.9 per cent defense work using our normal line of products. A few examples are: industrial air conditioning for production of munitions, vital chemicals, synthetic materials, control of atmospheric conditions in producing precision instruments, engines, et cetera; refrigeration for naval and other ships, cantonments, et cetera. (Appendix III gives more detail.)

In addition, those of the companies in our industry which have suitable facilities have taken substantial contracts for special defense work such as artillery pieces, gun mounts, machine guns, field kitchens, sheet metal work, and numerous other items.

The AC&R industry produces machinery, parts, fixtures, and accessories for applications ranging from large installations involving thousands of horsepower, all the way down to small machines of fractional horsepower. All sizes and types have some applications for defense requirements. This large range of types and sizes naturally involves production facilities of great variety.

Essential to Country

The AC&R industry is thought of at times, by some people, as being largely concerned with producing equipment of the non-essential—or even luxury—class, easily done without in times such as the present. Only to a small extent is this true, because by far the larger part of the industry's normal peacetime production is devoted to essential uses for the basic life of the country. In wartime these essential civilian uses are the foundation for the defense program itself. For example: Refrigeration is essential for the processing, transportation, and distribution of foodstuffs, providing palatable drinking water in hospitals and factories, producing synthetic rubber, et cetera; air conditioning increases the quantity and quality of blast furnace iron production and is essential for processing textiles, chemicals, certain medicines, et cetera. (Appendix III gives more detail.)

Although it is true that years ago this country thrived without the benefit of mechanical refrigeration and air conditioning, today the great urban concentration of people and modern production methods have developed ways of living and processes of manufacturing which are dependent upon refrigeration and air conditioning.

Many defense plants have been built,

TO MEMBERS OF THE INDUSTRY

Herewith a copy of the report to which you contributed statistical data and which the committee, on behalf of the air conditioning and refrigeration industry, has presented to the Office of Production Management.

In the preparation of this report, the committee made every effort to present a fair and realistic appraisal on the basis of the facts available. The consensus of the committee is represented in the conclusions reached and the statements made. While not all members of the industry may agree with some parts of the report, we are confident that you will realize that our task was not easy. Each member of the industry will undoubtedly suffer in some way. However, it is certainly better for each of us to be hurt a bit than for us all to be crippled beyond recovery—in which event we could neither do our direct defense job nor take care of really essential civilian uses.—OPM Subcommittee No. 4, Priorities, Ratings, and Allocations, Air Conditioning and Refrigeration Industry (Industrial and Commercial), James A. Bentley, W. C. Allen, F. H. Faust, C. V. Hill, Jr., M. W. Knight, L. C. Love, P. A. McKittrick, R. H. Pearse, A. B. Schellenberg.

and more are in process of construction, at points where there has been little population in the past. Housing is being provided by governmental and other agencies; but to serve these new concentrations of people, stores and markets must follow and refrigerating and air conditioning equipment must be made available. If proper living facilities are not provided, population shifts are difficult, costly, and delayed. Equipment for such installations is of the same importance to national defense as equipment for the training camps, which now carries preference ratings. To quote from a letter from the head of a large dairy products company:

"The importance of milk in the diet of both the civilian and military population of this country engaged in all-out defense effort is being repeatedly emphasized by government sources. We are supplying not only workers in civilian industries but camps of the U. S. armed forces with large quantities of fluid milk. We are receiving increasingly large orders for deliveries of evaporated and dried milk to United States and British governmental agencies. However, in the face of this increase in demands on our productive capacity, we are now encountering difficulties in obtaining refrigerating machinery because of the priority situation with respect to many of the materals necessary for the fabrication of this equipment."

A large number of AC&R factories are situated in small cities or towns, and a substantial proportion of the residents of such communities directly or indirectly earn their livelihood from these factories. A major portion of the workers employed in manufacturing, installing, and servicing the products of our industry are highly-trained specialists in their particular field.

We now have no means of obtaining quantities of materials, such as copper and steel, needed for the essential civilian uses. Members of our industry have been exerting considerable ingenuity toward developing substitutes for the scarcer materials. Efforts in this direction will continue; however, technical considerations definitely limit the extent of such substitutions.

We do not claim that all uses of our products are necessary during wartime, and we feel that unnecessary uses should be curtailed. We want to be realistic; we have voluntarily made sacrifices and will gladly make further sacrifices where advantageous to our country. However, we maintain that the best interest of the defense program, and the basic life of the country, will be served by enabling us to obtain needed materials for essential uses.

The requirements of the navy, army, maritime commission, and other governmental agencies for our industry's products will continue to be not only of very large volume but also of the greatest urgency as to speed of production and delivery. Unfortunately, we are able to anticipate only a small part of this need by building equipment in advance, because it is not possible for the abovementioned departments of the government to give us, in advance, the detailed information that is essential before we can place much of this highly-technical equipment in production. The result is that we are given information and orders in spasmodic rushes—and because many vitally-important things hinge on the speed of production and delivery we are able to attain, it is necessary for us to throw our organizations and facilities, overnight, into each emergency.

Highly Skilled Industry

It is impossible to expand and contract, at will, industrial organizations such as those represented in the AC&R industry which are, in the main, composed of highly-skilled engineers and mechanics of many years' specialized experience. The same statement applies, of course, to special tools and manufacturing facilities.

Our production of equipment for essential civilian uses would have the further advantage of enabling us to retain our skilled and technically-trained personnel; thus we would be able to do a better job on our direct defense work.

Now, however, we are faced with the paradox of the government's making it impossible for us to do the very thing which the government itself most desires and needs us to do.

The Proposed Plan

The Office of Production Management has invited us to make suggestions. Therefore, we propose the following simple plan, which should be effective in minimizing the use of critical materials and yet be relatively easy to administer:

Appendix II lists uses of refrigerating and air conditioning equipment for each basic industry of the country. Appendix III explains, in more detail, for just what purpose the equipment is used. Since there are so many different applications of air conditioning and refrigeration (industrial and commercial), there are necessarily many sizes and types of standard equipment produced, as well as special

Appendix I—Statistical Information Concerning Air Conditioning and Refrigeration Industry (Commercial and Industrial)

I.	Persons employed at August 31, 1941	
	(a) Factory and office employees	34,825
	(b) Field personnel: engineering, construction, service, sales	86,365
	(c) Secondary employment required to provide for item V estimated as	42,000
	(d) Approximate total employment	163,190
II.	(a) Annual payroll for I(a)	$ 58,227,666
	(b) Annual payroll for I(b)	140,315,650
	(c) Annual payroll for I(c) estimated as	58,800,000
	(d) Approximate total payroll	$257,343,316
III.	(a) Manufacturers' total orders booked, 12 months ending August 31, 1941	$202,784,164
	Note: This represents only the manufacturers' selling price of factory products, with the exception of certain complete installations made by some manufacturers.	
	(b) Additional costs, including local labor, locally-purchased materials, freight, etc., and installers' profit, not reflected in III(a) or V; estimated at	$364,176,246
	(c) Total installed cost to users for 12 months ending August 31, 1941, estimated at	$566,960,410
IV.	(a) Total manufacturers' unfilled orders at August 31, 1941	$ 63,011,301
	(b) Percentage of (a) covered by DEFENSE ORDERS	39.9%
V.	Total purchases by manufacturers from others of raw materials, fabricated and semi-fabricated parts, for 12 months ending August 31, 1941	$ 94,962,815
VI.	Floor space occupied (sq ft)	15,350,134
VII.	Quantities (in tons) of principal materials used during 12 months ending August 31, 1941:	
	(a) Steel	102,457
	(b) Iron	42,567
	(c) Copper and copper alloys	14,153
	(d) Brass	6,310
	(e) Aluminum	1,671
	(f) Nickel and nickel alloys	1,193
	(g) Lead	847
	(h) Zinc	794
	(i) Tin	783

Note: This information has been compiled from confidential data supplied for 163 companies. It covers practically all companies, both large and small, in all branches of the air conditioning and refrigeration industry (commercial and industrial). Returns were received from all questionnaires sent out, with only two unimportant exceptions. It may be conservatively stated that the figures are better than 98 per cent complete. The figures are factual with the exception of those stated as estimated. Such estimates as have been made are conservative.

Appendix II(a)—Air Conditioning Equipment

[Note—Appendix II(b), not given here, covers refrigerating equipment]

ESSENTIAL CIVILIAN USES

CHEMICALS
(Manufacturing Processes*)
COMMUNICATIONS
Telephone Apparatus (Manufacturing Processes*)
Telegraph Apparatus (Manufacturing Processes*)
Telephone Exchanges (Atmospheric Control)
DRUGS AND PHARMACEUTICALS
(Manufacturing Processes*)
ELECTRIC POWER (Manufacturing Processes* for Electric Motors and Other Electrical Equipment)
EXPLOSIVES (Manufacturing Processes*)
FOODSTUFFS
Processing of Foodstuffs*
Flour Processing*
Baking Processing*
Cereals Processing*
Miscellaneous Foods Processing*
Distribution of Foodstuffs
Preservation, Storage, and Sanitation Equipment for
Retail Stores
Eating Places
HOSPITALS AND INSTITUTIONS
Control of Operating Room Atmospheric Conditions
Control of Convalescing Space Atmospheric Conditions
HOUSING
Industrial
Factories
Drafting Rooms
Laboratories
Institutions
Hospitals
LABORATORIES
MACHINES AND MACHINE TOOLS (Production of)
Abrasives Processing*
Precision and Tolerances of Parts Control
METAL WORKING—FERROUS
METAL WORKING—NON-FERROUS
METALS
Mining
Processing*
Iron and Steel Production (Blast Furnaces, etc.)
OPTICAL APPARATUS (Processing* of Lenses, etc.)
PRECISION INSTRUMENTS (Manufacturing Processes*)
SYNTHETIC PRODUCTS
Plastics
Unclassified
TEXTILES AND CLOTHING (Manufacturing Processes*)
Non-Synthetic
Yarn Spinning
Fabric Weaving
Garment Making
Hosiery Making
Parachute Making
Other Fabric Production
Synthetic
Base Materials Manufacture
Yarn Spinning
Fabric Weaving
Garment Making
Parachute Making
Other Fabric Production
TRANSPORTATION
Motors and Engines Production
Airplane Production
Truck Production
Ships and Boats

LESS ESSENTIAL CIVILIAN USES

COMMUNICATIONS
Radio Tubes and Apparatus (Manufacturing Processes*)
Broadcasting Studios (Atmospheric Control)
Printing and Publishing (Processing*)
FOODSTUFFS
Processing of Foodstuffs*
Sugar
Candy Processing*
HOUSING
Commercial
Stores
Offices
Hotels
Public
Auditoriums
Public Buildings
Unclassified
Institutions
Unclassified
LEATHER AND LEATHER PRODUCTS
PAPER AND PAPER PRODUCTS (Manufacturing Processes*)
PHOTOGRAPHIC MATERIALS (Manufacturing Processes*)
Film
Cameras
TRANSPORTATION
Automobile Production

NON-ESSENTIAL USES

BUILDING MATERIALS AND SUPPLIES (Manufacturing Processes* for Cement, Ceramics, Glass, Lumber, etc.)
ELECTRICAL APPLIANCES (Manufacturing Processes*)
FLOWERS (Growing and Sale of)
HOUSING
Residential
Homes
Apartments
Commercial
Banks
RECREATION
Theaters
Bars
Clubs
Unclassified
TOBACCO PRODUCTS (Manufacturing Processes*)

*"Processes" or "Processing" denotes operations for which equipment is essential for purposes of controlling chemical, biological, or other reactions of given hygroscopic and non-hygroscopic substances, materials, or products. Whenever improved efficiency, health, and comfort of workers also results, this is purely an incidental additional advantage.

designs to meet particular needs. Because of the varying demand for any one size for essential and non-essential uses, it would be impracticable to curtail production by means of a quota based on sizes or types. It should be relatively easy, however, to decide that certain basic industries which require air conditioning or refrigeration are essential or unessential in the present emergency, and then to determine that the function of air conditioning or refrigeration is either necessary or not necessary for the particular industry. In Appendix II, we have indicated our objective opinion as to the degree of essentiality of each use.

a) The Office of Production Management would determine those civilian uses of air conditioning and refrigeration which are "essential," "less essential," and "non-essential". The method of making this determination might be along the lines which we have suggested in Appendix II.

b) A blanket rating (A-3 is suggested) that is high enough to permit the obtaining of materials, such as copper and steel, would be assigned to our industry for the "essential" civilian uses.

c) A blanket rating of A-3 might also be assigned for the obtaining of materials to anticipate production requirements for "defense uses", limited simply to the minimum inventories needed for production schedules. This would eliminate the requirement of the complicated PD-25 defense supplies rating, which has proved almost impracticable for our industry. The production of equipment for direct defense uses and the production of equipment for essential civilian uses should proceed together, in order that production schedules may be arranged efficiently. The manufacturer should be able to obtain materials to anticipate production requirements for both, under the same supplies rating plan. A manufacturer cannot operate effectively unless he knows how much material he is going to get and when he will be able to get it.

d) A lower blanket rating might be assigned to enable the obtaining of more limited quantities of materials for those uses determined as "less essential" civilian uses.

e) A preference rating order would be issued for the granting of preference ratings needed to obtain supplies to anticipate production requirements for the filling of orders for the "essential" and "less essential" civilian uses and "defense uses."

f) For those uses determined as "non-essential," as suggested in Appendix II, no preference rating would be given.

g) Our industry would undertake the responsibility of closely supervising and of policing the uses of such materials, in conformity with the law. Further, it would, by affidavit, guarantee that materials obtained under the respective preference ratings would be applied only to the authorized uses.

Because of the great variation of equipment and types of applications covered by our industry, any curtailment formula for our industry would, at best, be extremely complicated and probably impracticable. We believe that the above program would obviate the need for a curtailment formula.

It is estimated that such a program would result in a curtailment of between 35 per cent and 50 per cent in the total production of our industry for civilian uses. Based on the production of our industry for the 12 months ending August 31, 1941, and allowing for increased demands for "defense uses," there should be a net reduction of at least 20 per cent in raw materials required

We are certain that we can, effectively and legally, control the use of our products, and we will gladly provide any reasonable guarantees as to usage which may be required.

Copies of this letter are being sent to those in OPM with whom we have discussed various aspects of our industry problem.

We respectfully ask for action. What we need more than anything else is a tangible program which will enable us to keep our skilled personnel together and allow us to produce equipment for essential civilian and defense requirements.

Very truly yours,
OPM Subcommittee No. 4

Appendix III—Typical Industries Involved in Defense or Essential Civilian Requirements for Which Refrigerating and Air Conditioning Equipment Is Necessary for the Purposes Shown (Partial List)

INDUSTRY	PROCESS INVOLVED	ESSENTIAL FOR
Abrasives	Processing and drying of carborundum wheels and paper.	Uniformity of product. Production scheduling.
Airplanes and Engines	Rivet temperature control. Engine and parts machining. Propeller processing. Paint and dope application. Assembly.	Controlled precision and uniform tolerances. Elimination of effects of humidity and temperature variations in paint and dope processes, and in laboratories and test rooms.
Ammunitions	Powder and explosives processing. Ballistics, detonators, fuses, ignitor packing, pompom mix. Grinding, priming, pressing, solvent reclaim. Tetryl blending and pressing.	Fixing rate of burning of powder Accurate weighing. Precision of parts. Static and humidity control. Safety from explosions.
Bakeries	Flour Storage. Fermenting. Dough mix. Proofing. Cooling.	Mould prevention. Control of fermentation. Uniform quantity production. Sanitation. Storage.
Cereals, Foodstuffs	Processes too numerous for listing.	Spoilage prevention. Uniform quantity production. Sanitation.
Chemicals	Processes too numerous for listing.	Control of rates and limits of chemical reactions. Quality and purity control.
Dairy Products	Processes too numerous for listing.	Uniform products, sanitation, preservation as otherwise impossible.
Electrical Products	Processing of motors, controls, radio parts, telephone equipment.	Precision, tolerance control, dust prevention, prevention of damage by humidity and perspiration action.
Films	Coating, drying, cutting, and packing.	Uniform pliability, thickness and coating. Scheduled quantity production, and safety from fire.
Food Preservation		Impossible otherwise to prevent spoilage.
Glass	Intermediate steps in processing.	Uniform quantity production with controlled schedules. Imperative in making non-shatterable glass.
Hosiery	All stages of processing.	Static prevention, uniform product, and production rate.
Instruments (Watches, Navigation and Fire Controls, etc.)	Tool processing. Parts processing. Gaging.	Precision, condensation prevention, tolerance control, dust elimination, uniform quantity production.
Laboratories	Industrial testing. Animal assay. Growth and sprouting. Crystallization.	Without controlled conditions to eliminate effects of variations, much of this work would be impossible.
Linoleum (Battleship)	Intermediate steps in processing.	Uniform product and schedules.
Machine Shops	Precision rooms.	Close tolerances, constant measurements.
Mines	Mining areas.	Maximum production.
Meats, Lards, etc.	Processes too numerous for listing.	Mould prevention, sanitation, ingredient control, curing time control, uniform quantity production, storage, preparation, etc., as otherwise impossible.
Opticals, Sights, etc	Lens development. Lens assembly. Cementing. Precision room. Experimental room.	Precision, tolerances, mar prevention, condensation prevention, etc., as otherwise impossible.
Pharmaceuticals	Processes too numerous for listing.	Uniform quantity production of many drugs would be otherwise impossible.
Petroleum Products	Various stages of processing. Testing laboratories.	Uniform quantity production. Octane requirements.
Plastics	Intermediate stages of processing.	Reaction control, uniform production.
Paints and Lacquers	Pigment processing, etc.	Control of air moisture needs, drying rate, etc.
Paper	Processing, testing, storing.	Control of strength, sizing, and moisture content.
Rubber (See Synthetic Products for Synthetic Rubber)	Drying and Curing. Latex processing. Roller cooling.	Process, schedule, and quantity control.
Printing (Maps, etc.)	Processing and storage.	Alignment control, static elimination, uniform sizing, production rate.
Steel	In blast furnaces.	Increased quality and quantity of production of iron with lower coke consumption.
Synthetic Products (Rubber, Nylon, Rayon, etc.)	Processes too numerous for listing.	These products could not otherwise be uniformly produced at all in quantity.
Telephone Exchange	Apparatus room. Switchboards.	Control of static, prevention of sparking, elimination of corrosives in air.
Textiles	Spinning. Weaving.	Static control, uniform texture and quantity production, otherwise impossible.
Transportation (Perishables)		Long hauls otherwise impossible.

Protecting Piping and Tanks from Cold Weather Damage

Siegfried Ruppright, Professional Engineer and Consultant, Outlines Method of Preventing Damage in Freezing Weather

COLD WEATHER is on the way; plant piping men should give consideration to precautions to be taken to prevent damage to exposed piping and tanks from freeze-ups. . . . The author tells here why there is a problem, and briefly describes use of electrical heating to overcome it

THE detrimental effect of cold weather on exposed piping and vessels can be one of many. The most frequently encountered and most painfully remembered is that of congelation to the point of stop up which is usually accompanied by rupture of the wall, be it that of the blood vessels in a frozen ear or of a water main which affects an entire community.

Precipitation of matter in suspension—dyes, for example—is readily caused by a drop in temperature. It takes much more than restoration of the appropriate temperature to bring about the previous state.

Crystallization occurs in saturated solutions as soon as the temperature drops. Even if the solution is kept in motion, screens, valves and other restrictions in area get plugged up, a serious matter in a chemical factory. In gaseous fluids condensation takes place (such as of water in coal gas) which causes at least rust to form but also collects where it is least wanted, causing all sorts of trouble. Essential constituents are often condensed from gaseous mixtures in cold weather, bringing about substantial changes in the quality of the mixture at the point of delivery, not to speak of the obstruction by the trapped liquids. The mere increase in viscosity caused by unduly lowered temperature often necessitates preventive measures.

Anti-Frost Measures

The superficial observer usually has the impression that "the trouble could easily have been prevented," but this is not always so. There are three kinds of preventive measures: (1) Removal of the fluid. (2) Insulation. (3) Replacement of heat loss, our subject here.

The first seems the simplest, but what it involves in most cases where danger from low temperature exists is more than sufficient to rule it out completely. A sprinkler system in a warehouse cannot be emptied, nor can a vat holding expensive dyes, nor a water or gas supply line.

The second measure, though it involves a first cost (and some maintenance in most cases), is often cheaper in the long run, and it can be applied at a convenient time once and for all. Generally speaking, it can be said that artificial and natural insulation retard heat flow and can delay the incurrence of damage by frost. How much delay is sufficient must be ascertained from case to case, as must how much insulation is required to effect that delay.

Heat Loss Replaced

As an emergency measure, the heat loss from the endangered piping or containers can be replaced by continuously replacing the chilled contents with warmer stock. A simple case of this kind is that of a water main naturally insulated by being placed sufficiently deep in the ground but tapped on top for a minor connection. Not only is the highest point of this connecting line closer to the surface and therefore less insulated, but because of its smaller size it will freeze up more rapidly than the main. Something must be done, and since he has no choice the user usually lets the water run continuously during cold weather, which is wasteful.

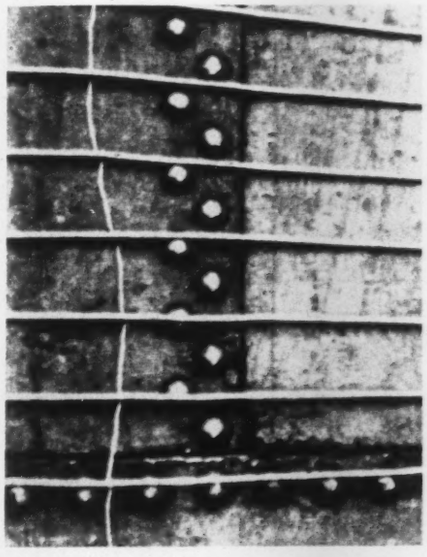


Fig. 1—Single strand copper wire applied to a large tank to replace heat loss electrically. The vertical tie wire serves to keep the electrical wire properly spaced

Electricity offers an excellent means of furnishing heat to replace the loss incurred at the wall of a container or pipe. On first sight, the wall of a metal pipe suggests itself for direct application of the current because it is a conductor of electricity, but its use may not be feasible for two reasons, even if it has no threaded joints and no electrolytic contents. The minor of these is that its resistance is of an order requiring a low voltage-high amperage relation which is expensive. The major is that a return conductor must be installed of a resistance negligible compared to that of the pipe in order to confine the heating effect to the pipe.

Usually feasible wherever danger from frost threatens, or a drop in temperature is not permissible for some reason or other, is the application of suitably selected wires between wall and insulation. Ordinary single strand copper wire frequently may be used; its application

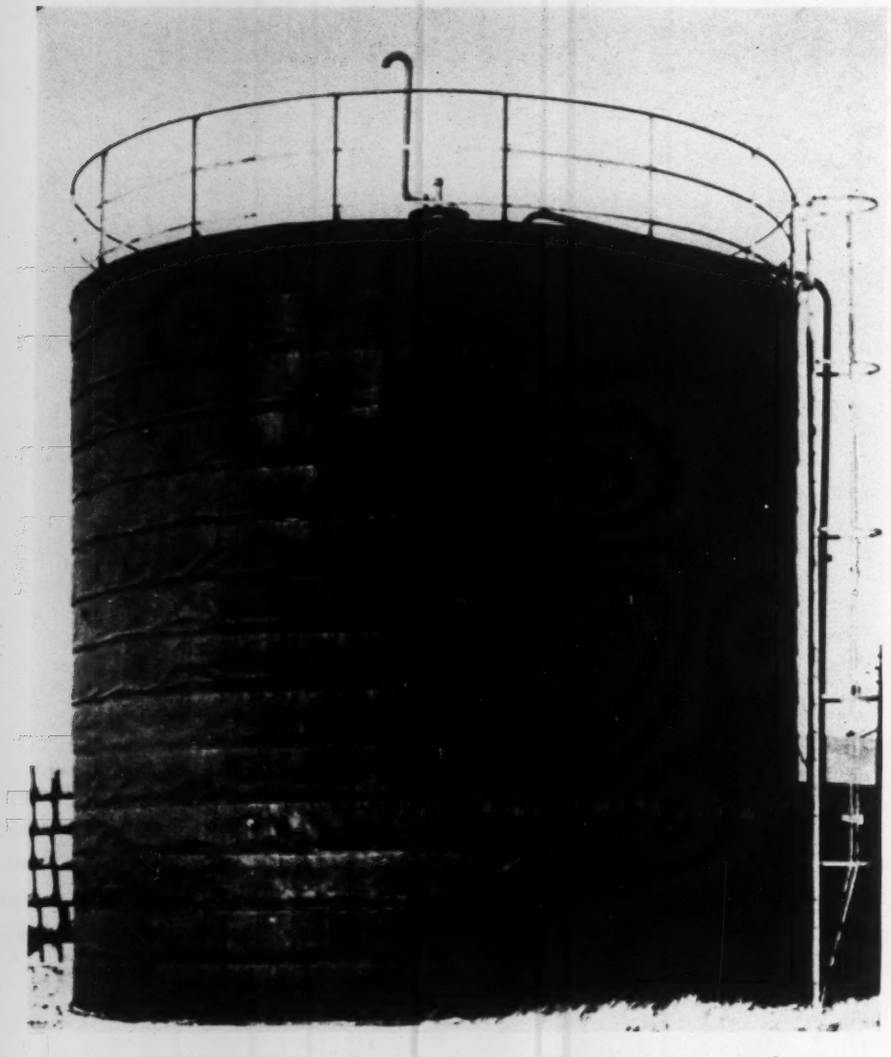


Fig. 2—Acid container at a chemical plant wired for 10 kw which will be consumed when outdoor temperature drops to — 13 F. No current is used at 32 F

on a large tank is shown in Fig. 1. The vertical tie wire serves to keep the electrical wire properly spaced.

Fig. 2 shows the finished container which holds an acid for a chemical factory. It measures 23 ft in diameter and is 25 ft high. It is wired for 10 kw which will be continually consumed when the temperature drops to —13 F, with no current used at 32 F.

Practical wattage for pipe lines is 4.5 watts per foot for 2 in., 7 for 4 in. and 11 for 8 in. pipes, according to my experience (although this varies with conditions); this is a negligible expense since it is incurred only for short spaces of time.

Heat Repels Dirt

IN the 1880's British scientists were puzzled by the fact that a hot surface remains clean in a dusty atmosphere, while a cold surface gathers dust. Where pipes at various surface temperatures run through a warm room, or in a building without uniform insulation, pipes, such as cold water conduits, at less than room temperature soon acquire dirt to which warmer pipes seem immune. Grime accumulates on walls and ceilings in streaks and patterns which outline cool areas caused by well-covered girders, laths and even nail heads. Indeed, one advantage of the panel heating system, in which the room and furniture surfaces are at higher temperatures than the air, is that those surfaces remain freer of dust.

Of recently revived importance to the problems of dust control, this ability of a warm surface to repel suspended particles is explained by the fact that the air molecules are in constant, rapid motion, moving in straight lines and rebounding from obstacles like perfectly elastic billiard balls. The molecules striking a surface warmer than themselves rebound on the average with greater speed and momentum than they had on striking, since they carry with them the energy the surface is losing by conduction. Or, if the surface is cooler than the air, the bounding molecules are slowed down.

About five years ago the British government introduced a dust sampling instrument utilizing this thermal repulsion which is said to remove from air all dust particles ranging in size from 20 microns down to the smallest particles visible in the ultramicroscope.

Although the use of thermal repulsion is now restricted to analysis, a recent discovery that a heated screen or lattice will act like a solid surface in repelling dust points toward the use of this principle for dust filtering. — *Industrial Bulletin of Arthur D. Little, Inc.*

FORMATION of a plumbing and heating defense industry advisory committee was announced October 27 by the bureau of clearance of defense industry advisory committees, Office of Production Management. Committee members are: George Hoffman, Crane Co.; William M. Byrd, Alabama Pipe Co.; E. S. White, U. S. Radiator Corp.; W. L. McGrath, Williamson Heater Co.; E. C. Sammons, Iron Fireman Mfg. Co.; J. A. Doucett, Revere Copper & Brass, Inc.; R. L. O'Brien, Detroit Brass and Malleable Works; O. L. Sweats, Grinnell Co., Inc.; M. W. Dennison, Braman Dow & Co.; L. F. Hudepohl, T. S. Conner, Inc.; H. M. Reed, American Radiator and Standard Sanitary Corp.; Charles D. Wessells, D. D. Wessells & Sons Co.; M. F. May, Young Radiator Co.; Frank C. Packer, Payne Furnace & Supply Co., Inc.; Harold Sweatt, Minneapolis-Honeywell Regulator Co.; N. H. Wiewel, Jones & Laughlin Steel Corp.; R. L. Stewart, Stockham Pipe Fittings Co.; G. E. Mumma, Sears Roebuck & Co.; and Edward Costello, Costello Engineering Co.

Paint Distribution Piping IMPROVES PRODUCTION

Berger's New Centralized Paint Storage and Mixing, With Nine Pipe Circulating System, Leads to Many Advantages in Manufacture of Metal Products

THROUGH nearly two miles of a new nine pipe circulating system, paint in as many different colors is now carried swiftly to the half a hundred points of actual application to shelving, steel furniture, kitchen cabinets, lockers and other metal products as they come down the production lines at the plant of the Berger Mfg. Div., Republic Steel Corp., Canton, Ohio.

Berger recently completed a sizable addition to its manufacturing facilities. A general relocation of departments, rearrangement of plant equipment and revamping of production lines was carried through at the same time and afforded the logical opportunity to go ahead likewise with a streamlining of the plant's paint handling methods.

Prior to the installation of the new centralized paint mixing and circulating equipment, enamels and lacquers had been delivered in 50 gal drums by factory trucks to four widely separated mixing stations situated near the various finishing areas in the plant. There, as required, the finishing materials were prepared for use by reducing with thinners, stirring by hand with wooden paddles for 10 to 15 min, etc. Spray operators had to come to these decentralized sources of supply on an average of once every 2 hr to refill the containers from which they sprayed.

Paint material mixed by hand in different locations, at different temperatures, at different times, and by not one but several individuals, could not be consistent in viscosity, specific gravity or degree of agitation—all of which have a direct bearing on the color uniformity to be obtained. Certain advantages to be derived from the centralized mixing of paints were, therefore, fairly obvious. Most important, continuous mechanical agitation could be used, a more complete dispersion of pigments obtained, and a homogeneous solution maintained at all times. Also, a more uniform temperature could be maintained in a mixing room than under any other conditions. Viscosity and specific gravity tests could always be made under the same conditions and would, therefore, be more directly comparable. Moreover, there could be undivided responsibility assigned to a single individual placed in charge of mixing operations on each turn. Handling by truck of paint materials throughout the factory could be eliminated. Finally, the fire hazard present when handling inflammable materials could be materially reduced.

Mixing Rooms

As finally evolved, two centralized mixing and paint storage rooms were established from each of which circulating systems fan out to several points on two floors of the plant. In the larger of the two mixing rooms there are 20 mixing tanks, 14 of which serve a group of seven paint circulating systems. Five of the seven lines carry paint to an equal number of spray booths in the first floor spray room. Two of the five lines continue on to a group of four spray booths on the second floor of the plant. A sixth paint line serves the second floor booths only. A seventh line runs to a group of dip enameling tanks on the first floor.

The smaller mixing room has nine mixing tanks from which two circulating systems carry paints to the locker assembly lines.

Together these two rooms provide storage facilities for more than 20,000 gal of enamels and lacquers, maintained at a year 'round temperature of 78 F.

The first floor spray room, with five lines serving each of five spray booths, is about 300 ft from the larger mixing room. Three of the four booths on the second floor, each served by three paint lines, accommodate two operators each, making a total of seven work stations involved. The longest distribution line, 1600 ft, runs from the main mixing room to the dip enameling area, where there are three outlets for three dip tanks.

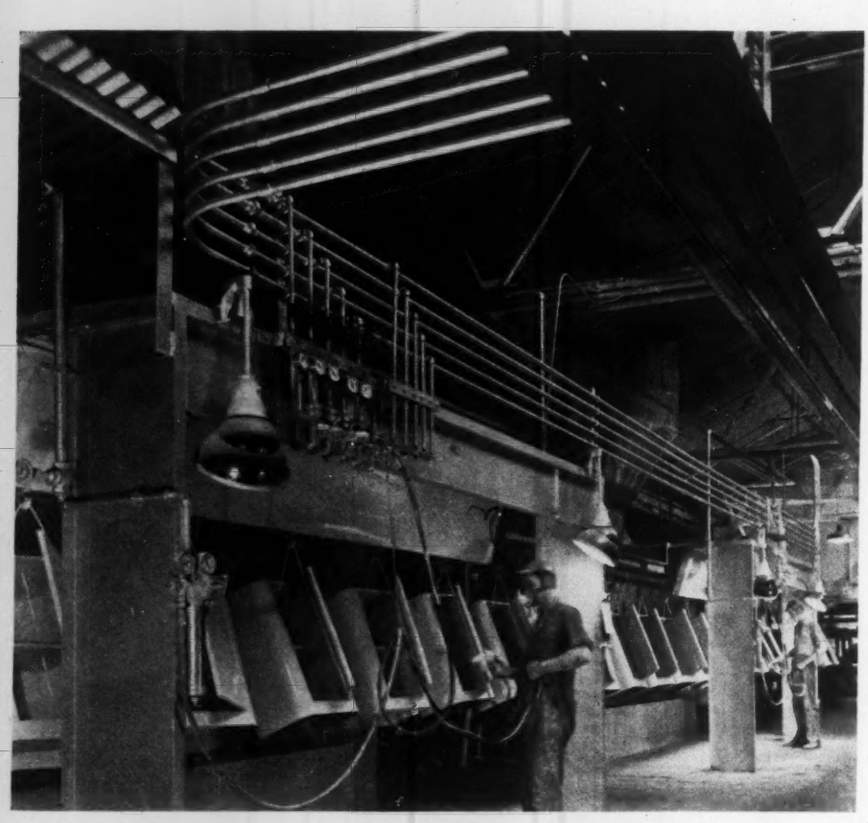
One inch pipe was used on all distribution lines having a total length of 1000 ft or more. Three-quarter inch pipe was used for all runs of less than 1000 ft in length. In every case the same size pipe was used for the entire length of a line. For all changes of direction, 12 in. minimum radius bends were used instead of pipe fittings.

The various pipe lines are currently being used to circulate the following colors for the following products, respectively:

Color	*Product*
Green Dipping Enamel	Shelving
White Synthetic Enamel	Roaster Cabinets
Gray Synthetic Enamel	Liquid Gas Cabinet
Green Synthetic Enamel	Furniture
Black Synthetic Enamel	Furnace Jackets & Sink Cabinet Bases
White Synthetic Enamel	Kitchen Cabinets
Ivory Synthetic Enamel	Automotive Display Cabinets
Olive Green Lacquer	Locker Door & Frame Assemblies
Brown Lacquer	Locker Door & Frame Assemblies

Back in the mixing rooms the circulating units which mix the

PAINT distribution is another way in which "industry's greatest transportation system" — piping — serves modern factories. The Berger Mfg. Div. of Republic Steel Corp. recently expanded its facilities and at the same time installed the new paint storage, mixing and circulating system which is described here



Regardless of the location in the plant, the paint is now sprayed under the same conditions of pressure, viscosity and specific gravity. This is especially important where parts such as these cabinet bases for electric roasters must match identically as to color in final assembly

paint materials and feed them into the distribution lines are made up of a pair of mixing tanks (one used for mixing, one serving as a reservoir), a pump unit, a filter and the necessary valves, gages and piping to and from points of use.

Tanks

Each tank is equipped with a mechanical agitation unit. A vertical shaft extending nearly down to the bottom of the tank has assembled on it two aluminum paddles of the propeller type. A ¼ hp, explosion-proof, gear head motor mounted on top of each tank is used to revolve the paddles at 59 rpm. Anti-whirl baffles are welded to the inside of the tanks to prevent swirling of the materials. The paddle blades are designed with an angle of pitch which tends to throw or lift the paint materials upward as they are rotated. In this fashion the heavier pigments that otherwise would tend to settle out of solution with the lighter solvents are kept thoroughly mixed.

Primarily there is no difference between the function performed by the larger group of 14 tanks, and the smaller group of six tanks along the opposite wall of the same room. They are all mixing tanks equipped with mechanical agitators. However, there is a difference in the way the tanks are used. The larger group is operated in conjunction with the seven pipe circulating systems for handling production colors. The smaller group is used for mixing batch quantities of paint required for short runs of special colors.

Each of the seven circulating systems in the larger mixing room is equipped with an internal gear type of pump driven by a 1 hp explosion-proof motor. These pumps deliver 5 gpm.

In the smaller mixing room compressed air is used to force the paint out into the lines. Small air-motor-driven internal gear pumps keep the liquids circulating. Less expensive than the others, this type of unit is satisfactory for short runs of pipe which, as in this case, have only one or two outlets.

In each pipe line system, just beyond the pump, is a mechanical filter. The circulation cycle is from the tank to the pump, through the filter, out into the line and return to the tank. Consequently the paint is continuously filtered as it is circulated and before it enters a distribution line.

To maintain uniform spraying pressure, each outlet in each pipe line is provided with a fluid regulator to reduce the line pressure. Each regulator has a removable key adjustment and a pressure gage for indicating the selected spraying pressure. Only the supervisor is provided with a key to operate the regulators. This centers in him the responsibility for always using the correct spraying pressure.

Many Advantages

There are several advantages of this paint circulating system used in conjunction with the centralized mixing and storage facilities. There is no longer any chance of dilution of the paint or of dirt getting into it, since it is now delivered to the spray guns exactly as mixed. With each outlet provided with an individual pressure control, it is possible to obtain and maintain a uniform spraying pressure which is also the correct pressure for the particular application of enamel or lacquer used. Regardless of location in the plant, the paint is now sprayed under the same conditions of pressure, viscosity, and specific gravity; this is particularly important where parts finished on different production lines are subsequently expected to match identically as to color in final assembly. Finally, due to the large storage facilities now available and the ability to maintain a uniform temperature of 78 F the year around, larger quantities of the paint materials can be purchased at one time, and the last step provided in insuring uniformity of color.

***OPM SURVEY**—A questionnaire covering the commercial refrigeration and air conditioning industries will be sent out shortly to ascertain where vital defense materials may be saved in those industries. The questionnaire is being prepared by the refrigeration unit of the electrical products and consumers durable goods branch of the division of civilian supply, office of production management, Washington, D. C.*

Among the products to be surveyed are unit air conditioners, condensing units, compressors, cooling coils, display cases and coolers, water coolers, beverage coolers, ice cream cabinets, reach-in refrigerators and beer cooling equipment. Information will be requested on inventory, production and materials, as well as distribution channels.

Reference List of Priority Orders

Following is the latest list of "P Series" and "M Series" priority orders of the Office of Production Management available for publication at the time of going to press. The related form numbers (the PD numbers) are also given:

"P SERIES"

P-1. Material for the production of electric traveling cranes assigns preference rating A-1-c. (Revoked by P-5).

P-2. Material for the production of machine tools. Assigns preference rating A-1-a. (Superseded by P-11). PD-6.

P-2-a. Material for the production of chucks. Assigns preference rating A-1-a. (Superseded by P-11). PD-6.

P-2-b. Material for production of spindles. (Superseded by P-11). PD-6.

P-2-c. Material for the production of hydraulic cylinders. (Superseded by P-11). PD-6.

P-2-d. Materials for the production of die heads. Assigns preference rating A-1-a. (Superseded by P-11). PD-6.

P-2-e. Material for the production of presses and air equipment. (Superseded by P-11). PD-6.

P-2-f. Material for the production of shell tapping machinery and die heads. Assigns preference rating A-1-a. (Superseded by P-11). PD-6.

P-2-g. Material for the production of air equipment. (Superseded by P-11). PD-6.

P-2-h. Material for the production of turret lathes. Assigns preference rating A-1-a. (Superseded by P-11). PD-6.

P-2-i. Material for the production of nibblers. Assigns preference rating A-1-a. (Superseded by P-11). PD-6.

P-2-j. Material for the production of gages. Assigns preference rating A-1-a. (Superseded by P-11). PD-6.

P-2-k. Material for the production of chucks, air, and hydraulic equipment. Assigns preference rating A-1-a. (Superseded by P-11). PD-6.

P-2-l. Material for the production of spindles, and hydraulic cylinders. Assigns preference rating A-1-a. (Superseded by P-11). PD-6.

P-2-m. Material for the production of machine tools. Assigns preference rating A-1-a. (Superseded by P-11). PD-6.

P-2-n. Material for the production of magnetic chucks and grinding machines. Assigns preference rating A-1-a. (Superseded by P-11). PD-6.

P-2-o. Material for the production of nibblers and cutoff machinery. Assigns preference rating A-1-a. (Superseded by P-11). PD-6.

P-2-p. Material for the production of tornio and automatic screw machines. Assigns preference rating A-1-a. (Superseded by P-11). PD-6.

P-2-q. Material for the production of drill braces. Assigns preference rating A-1-a. (Superseded by P-11). PD-6.

P-2-r. Material for the production of power-driven metal working machines and testing equipment. Assigns preference rating A-1-a. (Superseded by P-11). PD-6.

P-2-s. Material for the production of machine tool parts. Assigns preference rating A-1-a. (Superseded by P-11). PD-6.

P-3. Material entering into airframes. (Not including engines and propellers). Assigns preference rating A-1-d. PD-13.

P-3 Ext. Extended P-3 to Oct. 31, 1941.

P-4. Materials entering into airplane engines and propellers. Assigns preference rating A-1-c. PD-14.

P-4 Ext. Extended P-4 to Oct. 31, 1941.

P-5. Material for the production of cranes. (Corrected). Assigns preference rating A-1-a. Revokes Order No. P-1.

P-5. Amend. 1. Amendment No. 1 to general preference order P-5 for material for the production of cranes so as to include hydraulic bridge brakes in exhibit A. Assigns preference rating A-1-a.

P-5-a. Material for the production of cranes and hoisting equipment.

Defense Supplies

P-6. Defense supply rating order. Assigns preference rating A-10 to certain deliveries. PD-25. PD-25-b. PD-25-c. PD-25-d. PD-25-e.

P-6-a. Assigns A-10 rating to producers of civilian aircraft.

P-7. Material and equipment entering into merchant ship construction. Assigns ratings A-1-a, A-1-b, and A-1-c to ships in accordance with year of completion. PD-30. PD-30-a.

P-8. Material and equipment entering into freight car construction including railroad, industrial and mine freight cars. Assigns preference rating A-3. PD-38. PD-38-a.

P-9. Material entering into production of airframes for heavy bombers. PD-43. PD-43-a.

P-9-b. Material entering into production of aircraft engines for heavy bombers. PD-44. PD-44-a.

P-9-c. Material entering into production of aircraft propellers for heavy bombers. PD-45. PD-45-a.

P-9-d. Material entering into production of aircraft gun turrets for heavy bombers. PD-46. PD-46-a.

P-9-e. Material entering into production of aircraft gun fire controls and bomb sights. PD-47. PD-47-a.

P-9-f. Material entering into production of aircraft turbo superchargers for heavy bombers. PD-48. PD-48-a.

P-10. *Not issued.* PD-41.

P-11. Material for the production of metal working equipment. Assigns preference rating A-1-a, A-1-b, and A-1-c to items included in exhibit A. (Superseded by P-11-a). PD-42. PD-42-a.

P-11-a. Material for the production of metal working equipment. PD-81.

P-12. Preference rating order for aluminum scrap. Assigns preference rating A-10. PD-81-a.

P-13. Material entering into production of airframes. (Models SB-2-c and SO-3-c). PD-52. PD-52-a.

P-14-a. Material and equipment entering into the construction of shipways, which will produce completed ships during 1941. Assigns preference rating A-1-a. PD-56. PD-56-a.

P-14-b. Material and equipment entering into the construction of shipways, which will produce completed ships during 1942 and 1943. Assigns preference rating A-1-b. PD-56. PD-56a.

Electrical Relays

P-15. Materials entering into the production of electrical relays and solenoid assemblies. Assigns preference rating A-1-d. PD-57. PD-57-a.

P-16. Materials entering into the production of radio receiving, transmitting and directional equipment. Assigns preference rating A-1-c. PD-58. PD-58-a.

P-17. Material entering into the production of canning machinery and equipment. Assigns preference rating A-2.

P-18. Material entering into the production of cutting tools equipment. PD-6 revised.

P-18-a. Materials for the production of cutting tools. (Supersedes P-18). PD-81.

P-19. Material entering into the construction of defense projects. PD-63. PD-63-a.

P-19-a. Material entering into the construction of defense projects. PD-68.

P-19-b. Material entering into the production of defense projects. PD-68-a.

P-19-c. *Not issued to date.* Cancelled.

P-19-d. Defense project order.

P-20. Material entering into the construction of specified locomotives. PD-64. PD-64-a.

P-21. Material entering into the repair and building of steam, electric or diesel locomotives whether for railroad, mining or industrial use. PD-65. PD-65-a.

Repairs

P-22. Repairs.

P-22-a. Repairs and maintenance. (Superseded by P-22). Obsolete.

P-22. Interpretation—No. 1 to 3 inc.

P-23. Materials for the production of mining machinery and equipment specified herein.

P-24. Material entering into the production of experimental research work. PD-74. PD-74-a.

P-25. Material entering into the production of light tanks.

P-25-a. Material entering into the production of parts, accessories, and equipment for light tanks.

P-25-b. Material entering into the production of hulls, spare assemblies and spare parts for light tanks.

P-25-c. Material entering into the production of 30 caliber and 37 millimeter guns together with gun mountings for light tanks.

P-25-d. Material entering into the production of gasoline engines model R670 and diesel engines model T1020 and engine accessories for light tanks.

P-26. Material entering into the production of medium tanks.

P-26-a. Material entering into the production of parts, accessories, and equipment for medium tanks.

P-26-b. Material entering into the production of hulls, spare assemblies and spart parts for medium tanks.

P-26-c. Material entering into the production of 37 caliber and 75 millimeter guns together with gun mountings for medium tanks.

P-26-d. Material entering into the production of gasoline engines model R975 and diesel engines model T1400 and engine accessories for medium tanks.

P-27. *Not issued to date.*

P-28. *Not issued to date.*

P-29. Material entering into production of health supplies. PD-79.

P-29. Amend. Redefines "Health Supplies."

P-30. *Not issued to date.*

P-31. Material for the production of foundry equipment and repair parts. PD-81.

P-32. Material entering into production of parts of farm machinery and mechanical equipment. PD-88.

P-33. Material entering into the production of parts of farm machinery and mechanical equipment. PD-81.

P-34. *Not issued to date.*

P-35. *Not issued to date.*

P-36. *Not issued to date.*

P-37. *Not issued to date.*

P-38. Material entering into production of radiosondes.

P-39. Material for the production of arc welding and resistance welding machines. PD-81.

P-40. *Not issued to date.*

Defense Projects

P-41. Material entering into construction, maintenance and operation of defense projects.

P-42. Material entering into production of canning machinery and equipment. PD-6.

P-42-a. (Same). PD-6-a.

P-43. Material entering into the production of research laboratory supplies and equipment. PD-88. PD-93.

P-44. *Not issued to date.*

P-45. *Not issued to date.*

P-46. Utilities—maintenance, repair and supplies.

P-46. Amendment No. 1.

P-46. Interpretation No. 1 to 4 inc. PD-81.

P-47. Material entering into maintenance and repair of air transportation facilities. PD-96.

P-48. *Not issued to date.*

P-49. *Not issued to date.*

P-50. *Not issued to date.*

P-51. *Not issued to date.*

P-52. Material entering into production of aircraft accessories. PD-81.

P-53. Parts for the maintenance and repair of textile machinery and equipment. PD-81.

P-54. Material entering into the production of motor trucks, truck trailers and passenger carriers. (See L-1-a).

Defense Housing

P-55. Material entering into construction of defense housing. PD-105.

P-56. Materials for mine maintenance, repair and supplies. PD-119.

P-57. Replacement parts for passenger automobiles and light trucks. (See also L-4).

"M SERIES"

M-1. To direct the distribution of aluminum. PD-8. PD-26. PD-39. PD-40.

M-1-a. Schedule of preference ratings under M-1.

M-1-b. Modifies M-1 and M-1-a with respect to deliveries of low grade aluminum.

M-1-Ext. Extends orders M-1, M-1-a, and M-1-b to December 31, 1941.

M-1-c. To direct the distribution of aluminum scrap and secondary aluminum. Supplements orders M-1 and M-1-a. Repeals order M-1-b.

M-2. To direct the distribution of magnesium.

M-2-a. Supplementary order to M-2, distribution of magnesium.

M-2-Ext. Extends order M-2 to September 30, 1941.

M-3. To direct the distribution of ferro tungsten metal powder and tungsten compound. PD-9.

M-3-a. Supplementary order to M-3. Provides preference rating schedule under M-3. (Revoked by M-29).

M-3. Revised. Revised in form to meet requirements of federal register. (Revoked by M-29).

M-3-a. Revised. Revised in form to meet requirements of federal register. (Revoked by M-29).

M-3 and M-3-a Ext. Extends M-3 and M-3-a to August 31, 1941. (Revoked by M-29).

Neoprene

M-4. To direct the distribution of neoprene. PD-7. PD-36.

M-4-a. Supplementary order to M-4, distribution of neoprene.

M-4-b. Supplementary order to M-4 directing deliveries of neoprene.

M-4-c. Supplementary order to M-4 directing deliveries of neoprene in addition to supplementary orders M-4-a and M-4-b.

M-4-d. Supplementary order to M-4 directing deliveries of neoprene in addition to supplementary orders M-4-a, M-4-b and M-4-c.

M-4-e. Supplementary order to M-4 directing distribution of neoprene in addition to supplementary order M-4-a, M-4-b, M-4-c and M-4-d.

M-4-f. Supplementary order to M-4 directing distribution of neoprene during the month of May.

M-4-g. Supplementary order to M-4 directing distribution of neoprene in addition to supplementary order M-4-f.

M-4-h. Supplementary order to M-4 directing deliveries of neoprene in addition to supplementary orders M-4-f and M-4-g.

M-4-i. Supplementary order to M-4 directing deliveries of neoprene in addition to supplementary orders M-4-f, M-4-g, M-4-h.

M-4-k. Supplementary order to M-4 directing deliveries of neoprene in addition to supplementary orders M-4-f, M-4-g, M-4-h and M-4-i.

M-4-j. Supplementary order to M-4 directing deliveries of neoprene in addition to supplementary orders M-4-f, M-4-g, M-4-h and M-4-i.

M-4-l. Supplementary order to M-4 directing distribution of neoprene during the month of June.

M-4-m. Supplementary order to M-4 directing distribution of neoprene in addition to supplementary order M-4-l.

M-4-n. Supplementary order to M-4 directing distribution of neoprene in addition to supplementary orders M-4-l and M-4-m.

M-5. To direct the distribution of nickel bearing steel. (Revoked by M-21-a). PD-10. PD-11. PD-17. PD-18.

M-5-a. Supplementary order to M-5. Established preference rating and allocation schedules to users of nickel bearing steel. (Revoked by M-21-a).

M-5 Amend. Amendment to preference order M-5 providing definitions of certain terms. (Revoked by M-21-a).

M-5-a Amend. Amendment to supplementary order M-5-a establishing scope of preference order M-5-a. (Revoked by M-21-a).

M-5-b. Supplementary order to M-5. Defines nickel bearing stainless steel and percentage use of primary nickel to total nickel content. (Revoked by M-21-a).

M-6. To direct the distribution of primary nickel. PD-27.

M-6-a. Conserves supply and directs distribution of primary nickel. PD-27.

M-7. To direct the distribution of borax and boric acid. PD-31.

M-7-a. Supplementary order to M-7. Extends this order to expire July 30, 1941.

M-7 Ext. Extends this order to expire August 30, 1941.

Cork

M-8. To direct the distribution of cork and products and material of which cork is a component. (Continued by M-8-a. PD-28. PD-29.

M-8-a. To direct the distribution of cork and products and material of which cork is a component. PD-29.

Copper

M-9. To direct the distribution of copper. PD-37.

M-9 Amend. Amendment to preference order M-9 provides information on (1) pool (2) shipments, and (3) toll agreements.

M-9 Amend. Extends original order to include deliveries of copper base alloys and fabrication products of same.

M-9-a. Supplementary order to M-9 to conserve the supply and direct the distribution of copper. Revoked M-9 Amend.

M-9-b. Conserve the supply and direct the distribution of copper scrap and copper base alloy scrap. PD-1. PD-109.

M-9-c. Curtailing the use of copper in certain items.

M-10. To direct the distribution of polyvinyl chloride. PD-7. PD-36.

M-11. To direct the distribution of zinc. PD-50. PD-50-a.

M-11-a. Supplementary order to M-11 outlining amounts of metallic zinc, zinc oxide and zinc dust to be set aside each month. PD-62.

M-11 Amend. Amendment to M-11 providing information on: (1) shipments, (2) execution of PD-50 and PD-50-a, (3) toll agreements.

M-11-b. Supplementary order to M-11 outlining amounts of metallic zinc, zinc oxide and zinc dust to be set aside for August.

M-11-c. Supplementary order to M-11 zinc.

M-11-d. Supplementary order to M-11 zinc.

M-12. To conserve the supply, regulate processing, and direct the distribution of cotton linters. PD-36.

M-12 Ext. Cotton linters. Extends M-12 to July 31, 1942.

M-13. To direct the distribution of synthetic rubber. PD-7.

M-14. To direct the distribution and use of tungsten in high speed steel.

M-15. To direct the distribution of rubber and products and materials of which rubber is a component. PD-49.

M-15-a. Supplementary order to M-15, providing percentage of reduction in processing or consumption of rubber.

M-16. To conserve supply and direct use of tricesyl and triphenyl phosphates.

M-17. To conserve the supply and direct the distribution of pig iron. PD-71.

M-18. To conserve the supply and direct the distribution of chromium. PD-53-a. PD-53-b. PD-54. PD-55.

M-18 Amend. Chromium.

M-19. To conserve the supply and direct the distribution of chlorine.

M-20. To conserve the supply and direct the distribution of calcium silicon. PD-72.

M-21. To conserve the supply and direct the distribution of steel. PD-69, 70 and 73.

M-21-a. Alloy steel, alloy iron and wrought iron.

M-21-b. Steel warehouses. PD-83-a.

M-21 Amend. Steel. PD-73.

M-22. To conserve the supply and direct the distribution of silk.

M-23. To conserve the supply and direct the distribution of vanadium. PD-84.

M-25. To conserve the supply and direct the distribution of formaldehyde, paraformaldehyde, and hexamethylene tetramin.

M-25 Amend. Formaldehyde, etc.

M-26. To conserve the supply and direct the distribution of silk noils and garnetted or reclaimed silk fiber. PD-77.

M-26 Amend. Silk waste. Amendment to M-26.

M-27. To conserve supply and direct the distribution of phenols.

Refrigerant

M-28. To conserve the supply and direct the distribution of chlorinated hydrocarbon refrigerant.

M-29. To conserve supply and direct the distribution of tungsten. PD-9.

M-30. To conserve and direct distribution of ethyl alcohol and related compounds.

M-31. To conserve the supply and direct the distribution of methyl alcohol. (methanol).

M-32. To conserve the supply and direct distribution of potassium perchlorate.

M-33. To conserve supply and direct the distribution of potassium permanganate.

M-34. To conserve the supply and direct the distribution of toluene (toluol).

M-35. To conserve the supply and direct the distribution of phosphorus oxychloride.

M-36. To conserve the supply and direct the distribution of manila fiber and manila cordage.

M-37. To conserve supply and direct the distribution of rayon yarn. PD-102. PD-103-a. PD-103-b.

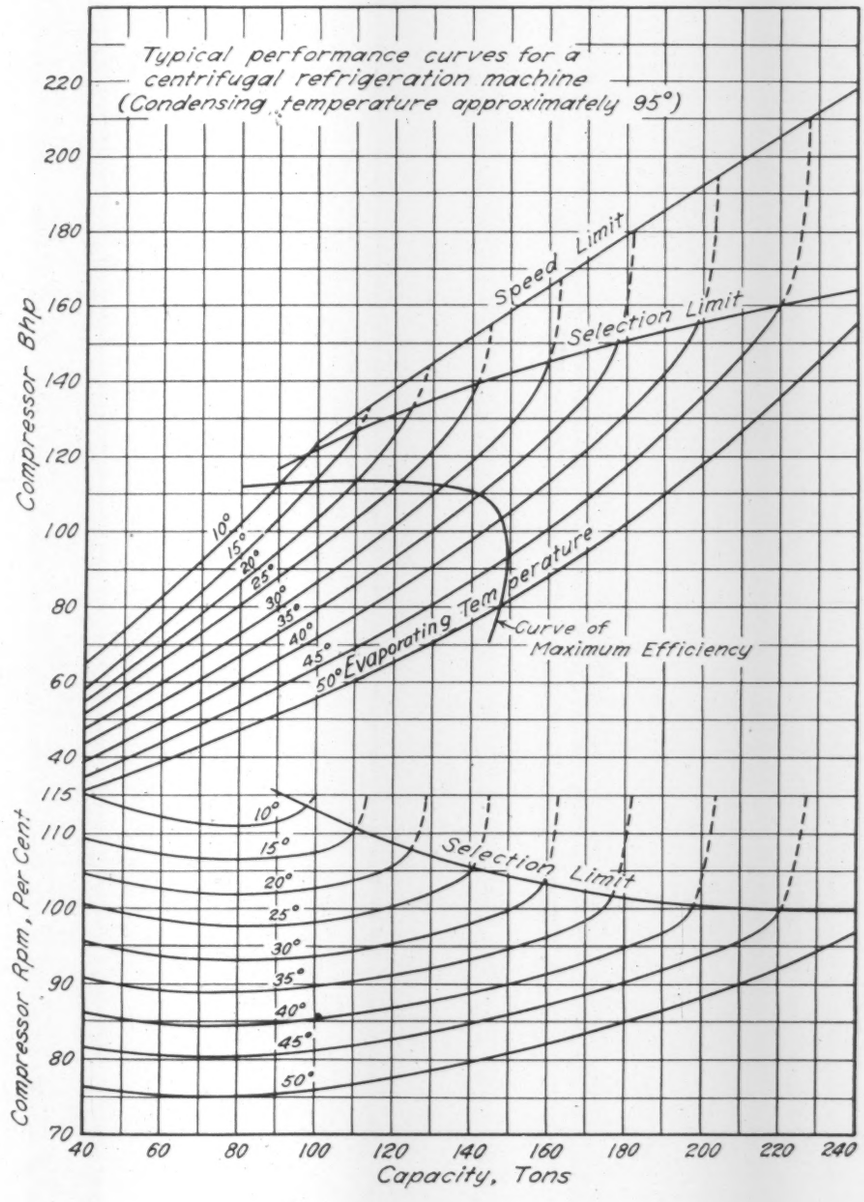
M-37-a. Supplementary order to M-37. PD-104. PD-112. PD-113.

M-38. To conserve the supply and direct the distribution of lead.

M-41. To conserve the supply and direct the distribution of chlorinated hydrocarbon solvents.

DOING A JOB FOR DEFENSE

Thousands of tons of centrifugal refrigeration compressors are on the job for air conditioning defense plants throughout the country, and for many process cooling applications. W. S. Bodinus, of the Carrier Corp., reviews briefly important characteristics of this useful type of machine. Operating head for a given constant speed changes very little for the largest part of the available capacity rating. Continued increase of load causes automatic unloading. Ratio of speed reduction to capacity reduction is small.



CENTRIFUGAL gas compressors are used in a manner quite similar to that of reciprocating compressors in refrigerating systems which operate on the normal compression cycle. The centrifugal compressor draws refrigerant vapor from the evaporator and compresses it until the vapor may be condensed when cooled. The refrigerants best suited for centrifugal compression are those of low pressure and high density with large volumes per ton of refrigeration.

The simple basic elements of one centrifugal machine are the shaft, impellers and casing, including the diffusers. The impellers and shaft rotate at high speed within a stationary housing. Supplementary elements of importance are a seal and a thrust bearing. The seal is required to prevent leakage of vapor where the shaft passes through the casing for connection to the motor or turbine drive unit. The thrust bearing carries unbalanced forces and holds the rotor in position.

Vapor enters the compressor axially, as guided by inlet vanes, and is moved at high speed by centrifugal force through the designed passages of the impeller. Discharged at the periphery of the impeller, the vapor passes through the diffusers in the casing and is delivered to the impeller of the next stage. The flow of vapor is continuous.

Pressure is acquired by the vapor in the impeller, as a result of the centrifugal force, and the vapor moving at high velocity also acquires kinetic energy. The diffusers, or the designed passages in the housing, provide for the conversion of this kinetic energy—the conversion of velocity to pressure. In consequence, the vapor is compressed to that pressure which is required for condensation in the condenser.

The ratio of compression is fixed by the temperature of evaporation and by the condensing temperature established by the available condenser water. The ratio of compression, or the temperature head of operation for air conditioning applications, may be satisfied by two stages of compression. For

greater temperature heads, as for commercial refrigeration, three stages (or perhaps more) are required. The number of stages for a given temperature head is determined by details of design, peripheral speed, impeller diameters and rotor speed.

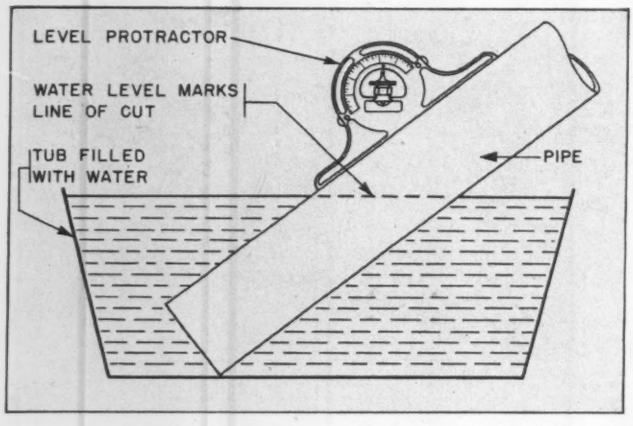
Capacity characteristics of the centrifugal refrigerating compressor are similar to those of centrifugal pumps. The curve which relates head to volume of gas pumped is comparatively flat for the major part of the range of capacity. For instance, the head may be maintained constant during the course of 50 per cent capacity reduction by only 3 per cent variation of speed. By virtue of this flat curve characteristic, the operating head for a given constant speed changes very little for the largest part of the available capacity rating. Near the maximum capacity, the curve turns downward, gradually and then very abruptly. The approach of an overload is accompanied by reduction of head and the continued increase of load causes automatic unloading. This is the result of the impeller design, a characteristic of backward curved impeller blades.

The flat characteristic is consequently of great value in relation to operation and performance. A machine driven at constant speed can operate very close to desired conditions when wide variations of load occur. Likewise, exact maintenance of conditions can be maintained by very little change of speed when the driving unit is provided with variable speed control. Very precise and satisfactory control of conditions can be simply accomplished by manual operation or automatic control devices. Since the ratio of speed reduction to capacity reduction is very small, the cost of speed variation with electric drives is also very small.

The centrifugal refrigerating machine is inherently one for comparatively large vapor volume. Accordingly, it is an ideal unit for large loads at ordinary temperature levels. No other type of system can provide so much refrigerating capacity per unit of space occupied or per unit of floor area. The centrifugal is also very suitable for low temperature operation because of the well known thermal characteristic that at low evaporating temperatures the volume of any refrigerant vapor per ton of capacity increases greatly. Hence the centrifugal not only fulfills requirements for "run of mine" applications but also has merit for new fields extending into large capacity plants and into low temperature operations.

Power requirements of the centrifugal compressor increase with capacity (not in direct proportion) up to a certain head similar to other centrifugal apparatus. This is the unloading characteristic previously mentioned. It is of practical importance because a compressor operating at constant speed and suction temperature will not overload the motor when the supply of condensing water fails. The rapid increase of head causes the compressor to quickly drop the load. When an installation is unusual and is such that a load in excess of the normal load can be imposed upon the unit, then the motive power should be selected for the peak condition.

How to Make Mitered Pipe Joints



There is an easy way to mark the correct line of cut on pipe when making mitered joints for pipe elbows. Sole requirements are a tub of water and a level protractor.

Submerge the end of the pipe under water at one-half the desired angle of bend, using a level protractor. By marking the pipe at the water level with chalk or soapstone, the correct line of cut for that particular angle is readily indicated. If desired, a templet can be made for duplicating the cut on other pipe of the same diameter, by placing a piece of heavy paper squarely about the pipe, tracing the cut, and then cutting away the scrap portion of the paper to form a permanent templet.

In the example shown in the sketch, the pipe is inserted in the water at an angle of 37½ deg which will give a two-piece mitered joint of 75 deg. If a bend consisting of more than two sections is desired, the correct angle of each cut required to arrive at the total angle of bend can be determined from the following equation: Angle of cut =

$$\frac{\text{Total Angle of Bend}}{2 \times (\text{Number of Sections} - 1)}$$

—W. D. Scheffey, service operator, The Linde Air Products Co.

Heating · Piping and Air Conditioning

AMERICAN SOCIETY of HEATING and VENTILATING ENGINEERS

Journal Section

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Entrainment and Jet-Pump Action of Air Streams

By G. L. Tuve,* G. B. Priester** and D. K. Wright, Jr.**
Cleveland, Ohio

This paper is the result of research sponsored by the American Society of Heating and Ventilating Engineers in cooperation with the Case School of Applied Science

Summary

WHEN air is discharged from an outlet into a free-open space or room, the primary air stream becomes a *jet pump* for moving the air surrounding it. Determining the behavior of such an air stream when it is entraining and mixing the room air is a problem in fluid mechanics which must be solved largely by experimental methods, although approximations may be made by analytical means.

In view of the increasing variety of grilles, slots, nozzles and venturi devices used for diffusion of the air supply in a room, a brief general survey of the subject of air diffusion by jet devices is presented. The air conditioning engineer is interested in the jet pump not only in connection with air streams in rooms, but also for such applications as fume exhausters, gas and oil burners, smoke prevention jets and steam-ejector refrigeration. Hence this paper includes an examination of data and results reported by several investigators.

Experimental results are presented for air streams discharging into free open space from various types and sizes of outlets, at velocities up to 2400 fpm. Using an experimental method of integration, these results are shown to confirm the analysis by the theory of conservation of momentum. The total air entrained was in direct proportion to the distance from the discharge outlet, but the ratio of entrained room air to primary air was found to be almost independent of the velocity at the outlet face. For each foot of throw or travel of the stream, the actual volume of room air entrained was 0.3 to 0.7 times the primary air volume. This entrainment was increased by using a small outlet, a narrow slot, or a spreading grille.

*Professor of Heat-Power Engineering, Case School of Applied Science. Member of ASHVE.
**Instructor in Mechanical Engineering, Case School of Applied Science. Junior Member of ASHVE.

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The Problem

The jet pump has a large number of applications, but each of these is so specialized that no comprehensive study of the device is available. The familiar puffing of a locomotive, the operation of the gas burner on a kitchen stove and the movement of air by a desk fan are common examples of air entrainment on the jet-pump principle.

Jet-pump equipment is built in a wide variety of forms, but for a general analysis Fig. 1 may be used to represent most of them. The essential part of the apparatus is the *nozzle* or discharge outlet from which the jet emerges. If the mixture is to be pumped or directed into an outlet pipe, the venturi or mixer section is used, but the entrainment or mixing action will take place even though the venturi element is entirely omitted and the jet discharges into free space. The primary stream P may be any fluid, air, water, steam, gas, etc. The entrained or secondary fluid S is usually air or water, although any other fluid might be substituted.

The engineer is interested in two questions about the jet pump:

(1) What are the best design proportions or dimensions for the device?

(2) What are its performance characteristics?

Principle of Jet Pump

The essential feature of the jet pump is to obtain the maximum of velocity or kinetic energy at the nozzle discharge-face. For low-pressure air or gas, and for all liquids, the process of accelerating the fluid as it approaches the nozzle discharge is by a simple conversion from pressure energy to kinetic energy:

$$\frac{P}{w} = \frac{V^2}{2g}, \text{ or } V = \sqrt{2gh} \quad \ldots\ldots (1)$$

where the velocity V in feet per second and the fluid density w in pounds per cubic foot represent the conditions at the nozzle discharge face. P is the drop in pressure, pounds per square foot, from the upstream section to the nozzle face, or h is the equivalent head in feet of fluid flowing. As long as compressibility and velocity of approach are both negligible, this equation may be used to compute the velocity at the nozzle face for any fluid, gas, liquid or vapor. From the practical standpoint this process is merely that of increasing the velocity of flow by reducing the size of conduit. When the pressure-drop is large and the fluid is a gas or a vapor, the fluid expands as it approaches the nozzle outlet due to the reduction in pressure, and a still higher velocity is produced. With such a high-pressure nozzle, some of the internal energy of the fluid is also turned into kinetic energy and the calculation of velocity may be based on the

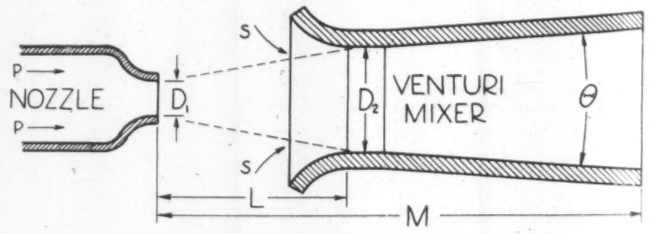


Fig. 1—Diagram of typical jet-pump apparatus

Table 1—Design and Performance of Jet Pumps for Air Entrainment
(*See Fig.* 1)

Application	Reference Footnote No.	Fluids		Design							Pressures: Jet Across Supply Mixer		Entrainment Ratio
		P	*S*	D_1	$\frac{D_2}{D_1}$	$\frac{L}{D_1}$	$\frac{M}{D_1}$	$\frac{L}{D_2}$	$\frac{M}{D_2}$	Angle θ deg	Psi	In. of Water	
Locomotive front-end	4	Steam	Gases	1.5	4.0	4	12	1	3	9	3.0	5	3.0
Fume exhauster	7	Air	Air	5.2	2.5	.....	36	.....	14	6	0.28	1.5	1.5
Fume exhauster	a	Air	Air	2.0	3.0	7	46	2.3	15	5	1.0	0	5.5
Fume exhauster	a	Air	Air	2.0	3.0	7	46	2.3	15	5	1.0	1.5	2
Gas burner	2	Air	Air	0.08	15	20	230	1.3	16	6	3.0	0	30
Gas burner	2	Air	Air	0.08	15	20	230	1.3	16	6	3.0	0.5	11
Gas burner	b	Nat. Gas	Air	.....	.....	.....	.....	2	8	4	.....	.....	.....
Experimental	5	Air	Air	0.38	3.0	15	.....	5	.....	16	0.9	0.5	3
Smoke preventer	c	Steam	Air	0.12	12	25	120	1.2	6	7	80	3.0	15
Experimental	6	Air	Air	0.50	1.5	8	25	5	16	16	0.9	6.0	2

[a]The Venturi Ejector, by F. F. Kravath. (*Heating and Ventilating Magazine*, June 1940, p. 17.)
[b]*U. S. Bureau of Standards Technologic Paper* No. 193, Design of Atmospheric Gas Burners, 1921.
[c]Design of Steam Jets for Smoke Abatement, by Du Perow and Bossart. (*Thesis*, Case School of Applied Science, 1927, 35 p.)

drop in enthalpy for a constant-entropy process.

But computing the nozzle velocity does not explain the pumping process. In what manner or by what means does this nozzle jet produce entrainment and mixing? There are in some cases at least four processes involved: (1) Acceleration of the particles of secondary fluid by impact of particles from the primary jet. (2) Entrainment of secondary fluid by viscous friction at the periphery of the primary jet. (3) Over-expansion of the primary jet to a pressure below that of the secondary fluid, with consequent flow of secondary fluid toward the axis of the jet. (4) Change of state, as in a steam injector, during which a large reduction in volume occurs by condensation, and the energy of latent heat is made available.

Theoretical analysis of a jet pump in which two or more of these processes are involved becomes very complicated. Hence the designs have been based largely on the results of experiments and tests. By plotting curves of pump efficiency, pressure and capacity, the effects of design changes can be studied.

Available Data on Jet-Pump Design

A brief discussion, with references, illustrates the possibilities of design and performance of the single-stage jet pump. Two classes of jet pumps might be recognized, *volume pumps* and *pressure pumps*. A similar classification is frequently applied to centrifugal pumps and fans. The volume pump is usually a large unit, or at least the venturi mixer is large as compared to the nozzle. The most common use of the pressure pump is as a vacuum pump, although the discharge pressure may occasionally be even higher than the jet-supply pressure, as in the locomotive injector. Most high-pressure pumps depend on the condensing of a vapor, and the venturi element is often compounded.

Table 1 shows typical design proportions for a number of single-stage pumps (see references). Most of these would be classified as volume pumps, but it is evident that even for similar applications there is a wide flexibility in the design. The diameter of the venturi throat is usually made 2 to 4 times the diameter of the nozzle. The venturi mixer must be accurately centered, with its throat at a distance of 4 to 20 nozzle diameters from the face of the nozzle. The venturi must be large enough so that the nozzle jet, which has a natural angle of around 20 deg, will not produce objectionable *splashing* at the inlet. The entrance to the venturi is either a rounded nozzle or a cone of about 20 deg angle followed by a short parallel section. The angle of the expanding diffuser beyond the venturi throat is usually between 4 deg and 10 deg and its length may be as little as twice its smallest diameter if necessary, although a longer diffuser is preferred for pressure recovery. Most investigators report improved performance of the pump as the diameter and length of the venturi mixer are increased, and the limitations are mainly those of space and cost. Some commercial devices use very small nozzles in multiple, or slot-type nozzles instead of round ones.

When the pressure difference across the venturi is increased, either by increasing the entrainment vacuum or by increasing the exit resistance, the pump becomes more expensive to operate because a much higher jet pressure is required. There is no easy way to calculate the jet pressure required for a given case. Trinks[1] has derived an equation for computing the nozzle jet velocity required to produce a specified pressure and velocity of the mixture at the diffuser outlet. His analysis was based on conservation of momentum and conservation of energy, with an assumed efficiency of pressure recovery in the diffuser of 50 per cent. The application was that of a gas burner mounted in the wall of a furnace, with a pressure of 0.025 in. of water in the furnace. Howe and Johnson[2] have used a similar analysis to derive a family of curves covering the range of nozzle supply pressures from 3 lb to 60 lb per sq in. gage, and for resistances to 2 in. of water. These curves were checked experimentally with reasonable success.

The pressure-volume relationships for a jet pump may be likened to the family of characteristic curves for a centrifugal fan. In both cases the shape of these curves, between wide-open and blocked-tight (S.N.D.) conditions, are affected by the design proportions. One of the authors has shown, by constructing and testing a series of jet pumps, that by changing the ratios D_2/D_1 and M/D_1 (Table 1), the curve of nozzle pressure against vacuum for a high-pressure air ejector may be widely varied to suit the application.[3] But several investigators[4,5,6] have reported that with the lower jet pressures (below 5 lb gage) and the usual design proportions, the

[1]Industrial Furnaces, by W. Trinks (Vol. 2, p. 54.)
[2]Atmospheric Type Burners Used with Natural Gas, by Howe and Johnson. (*ASME Transactions*, November, 1939, p. 673.)
[3]Design of Air-Atomizing Oil Burners, by G. L. Tuve. (*M. E. Thesis*, University of Minnesota, 1921, 55 pp.)
[4]A Study of the Locomotive Front-End, by E. G. Young. (*Bulletin University of Illinois*, Engineering Experiment Station, No. 256, 1933, 180 pp.)
[5]Efficiencies and Entrainment Ratios of Air Jet Pumps, by G. P. Miller. (*Master's Thesis*, Case School of Applied Science, 1940, 77 pp.)
[6]The Performance of Jet Pumps, by Herrick and Horsburgh. (*Thesis*, Case School of Applied Science, 1940, 40 pp.)

entrainment ratio is only slightly affected by changing the jet pressure.

For the design of fume exhausters, curves are given in Fan Engineering[7] which show the effect of changes in D_2/D_1 upon the entrainment ratio, the pressure and the efficiency. The accompanying text states that the pressure-capacity curve plotted over the range from static-no-delivery to full-open is substantially a straight line.

From the foregoing discussion of the single-stage jet pump with venturi, some conclusions may be drawn for application to the devices used for air mixing in rooms. If such a room air diffuser utilizes a venturi mixer, the entrainment within this mixer will probably be less than twice the primary air volume, due to the prohibitive size of the venturi for large entrainment. Entrained volume will be nearly proportional to the velocity of the primary jet (entrainment ratio constant), but the noise factor will limit the jet velocity which may be used. The entrainment might be increased by using small nozzles or narrow slots.

Jets in Free-Open Space

When a jet discharges into free-open space without a venturi mixer, the case is less complicated and is more easily attacked by theoretical analysis. Many such analyses have been reported, based on theories of diffusion or of turbulence, and involving various assumptions or approximations. Among the common simplifying assumptions are: (1) That the static pressure throughout the open jet is constant, *i.e.*, no vacuum exists. (2) That the viscous forces are small as compared with the turbulent or inertia forces. (3) That the primary and secondary fluids are of equal density. (4) That the jet is being discharged from a point source. (5) That the jet is entirely symmetrical. (6) That the velocity across the jet is uniform. These assumptions are all closely approximated in such a practical case as that of a small gas burner mixer, or in any low-pressure device where gas or air discharges from a small, circular, rounded-entrance nozzle into the atmosphere.

For the specific application of air discharged from an outlet opening or grille into free-open space in a room, the assumptions of a point source, jet symmetry and uniform velocity cannot be used. There is little error in assuming the pressure constant, *i. e.*, a negligible vacuum within the jet. Moreover, the viscous drag is small on account of the low viscosity of air, and the densities are practically equal. Re-examining the four mixing processes already enumerated, it is found that only one remains, that of impact of the primary stream upon the free air in the room. The theory of conservation of momentum is then applicable. The collisions of particles may be studied in accordance with the statistical theories of turbulence,[8] but since experimental verification and evaluation of constants are necessary in any case, it may be more direct to use a method of experimental integration instead of carrying out an analysis of the mechanisms of turbulent diffusion.

The mixing-space available with a free-open jet is greatly increased over that of a jet discharging to a venturi; hence a greater entrainment ratio might be expected. But there are few published data giving quantities of air actually entrained by open jets. That the entraining or mixing action is continuous along the path of the jet has been shown by tests of propeller fans. In fact, this principle is turned to commercial advantage when fans are rated by the *NEMA* Code.[9] A series of such tests made by the authors on six 4-bladed propeller fans from 24 in. to 50 in. wheel diameter showed an average rating of about 125 per cent of primary air volume at 12 in. in front of the fan, and about 155 per cent of primary air volume at 48 in. in front of the fan. One of the methods for increasing the entrainment of air by a simple jet is to use a slot-type outlet. The use of narrow slots for room air distribution has been discussed by Mackey,[10] Stevenson[11] and others.

One of the important quantities to be determined for a room-supply air stream is the total blow or throw of the primary jet. As a result of both analysis and experiment[10, 12, 13] Mackey and others have stated the total throw in terms of the area of the outlet and the primary air volume (or the face velocity):

$$T = \frac{KQ}{\sqrt{AB}} = KV\sqrt{AB} \quad \ldots\ldots (2)$$

where T is the total throw, Q the primary air volume, V the face velocity, and A and B the linear dimensions (assuming a rectangular outlet). Additional experimental results in support of this equation would be desirable.

The Momentum Theory

The mixing action of an open jet is due almost entirely to the impact process, as has already been indicated, and by simple application of the theory of conservation of momentum, many of the performance laws for a free-open stream may be developed. The assumptions are those already enumerated: Uniform density, pressure, temperature and

Table 2—Calculated Air-Stream Performance
(Assuming Uniform Velocity Across the Stream)

Outlet Area, Sq Ft	Outlet Size, In.	Aspect Ratio	Assumed Jet Angle, Deg	Outlet Velocity fpm	Total Throw ($V = 50$ fpm)	Total Entrainment (At $V = 50$ fpm)	Entrainment Per Foot of Stream, Per Cent
0.50	8.48 x 8.48	1	19	400	14.8	7	47.2
0.50	8.48 x 8.48	1	19	800	31.7	15	47.2
0.50	8.48 x 8.48	1	19	1200	48.5	23	47.2
0.50	12 x 6	2	19	400	14.7	7	47.5
0.50	12 x 6	2	14	800	42.9	15	35.0
0.50	12 x 6	2	19	800	31.6	15	47.5
0.50	12 x 6	2	22	800	27.2	15	55.0
0.50	12 x 6	2	30	800	19.7	15	76.0
0.50	12 x 6	2	60	800	9.2	15	63
0.50	12 x 6	2	90	800	5.3	15	83
0.50	24 x 3	8	19	800	30.6	15	49.0
0.167	4.9 x 4.9	1	19	800	18.3	15	82.0
0.167	24 x 1	24	19	800	16.6	15	90.5
0.083	24 x ½	48	19	800	11.1	15	135

[7]Fan Engineering, Buffalo Forge Co., 4th edition, p. 229.

[8]Investigations of Turbulent Mixing Regions Formed by Jets, by A. M. Kuethe. (*Journal of Applied Mechanics, ASME*, 1935, p. A85.)

[9]*NEMA* Fan Standards, Publication No. 39-56, *National Electrical Manufacturers' Association*, New York, N. Y., 1939.

[10]Air Conditioning Principles, by C. O. Mackey. (International Textbook Co., 1941, p. 184.)

[11]Air Discharge from Narrow Slots, by F. F. Stevenson. (*Heating, Piping and Air Conditioning*, May and June, 1941.)

[12]Air Paths from Grilles, by Greenlaw and Hart. (*Heating, Piping and Air Conditioning*, June and July, 1938.)

[13]Measuring Air Distribution and Grille Performance in Air Conditioning, by G. L. Tuve. (ASHVE Journal Section, *Heating, Piping and Air Conditioning*, November, 1937, p. 700.)

velocity, and negligible viscosity. The assumption of a point source is not necessary, and jet symmetry is assumed only with respect to the angle of spread, not to the opening itself. Among these assumptions, the most serious violation of practical conditions lies in the statement that uniform velocity exists across any section perpendicular to the stream, *i.e.*, at the outlet or parallel to it.

Constant momentum of the stream (MV = constant) means also that the total quantity of air in the moving stream is inversely proportional to the stream velocity. This is demonstrated as follows:

$$M_1 V_1 = M_2 V_2$$
$$wQ_1 V_1 = wQ_2 V_2$$
$$\frac{Q_2}{Q_1} = \frac{V_1}{V_2} \quad \ldots\ldots\ldots\ldots\ldots\ldots (3)$$

where Q is the total quantity or volume of air moving, at sections 1 and 2, V is its uniform velocity at that section, and w is its constant density. From this equation we can obtain the air entrainment at any plane where the velocity is known. If we arbitrarily define the end of the *throw* as that section where the average velocity has reached, say, 50 fpm, the total air in motion at the end of the throw for an outlet velocity of 1000 fpm, is 20 times the primary air volume. For an outlet velocity of 2000 fpm the total air mixture at the end of the throw would be 40 times the primary volume.

This does *not* mean that at a given plane, say, 10 ft from the outlet, the ratio of entrained room air to primary air increases as the face velocity increases. In fact, the entrainment ratio at a fixed distance from the outlet should be independent of the outlet velocity, as shown by the following derivation:

Since

$$Q = AV, \text{ or } V = \frac{Q}{A},$$

Equation (3) becomes:

$$\frac{Q_2}{Q_1} = \frac{Q_1 A_2}{A_1 Q_2}$$

or

$$\frac{Q_2}{Q_1} = \sqrt{\frac{A_2}{A_1}} \quad \ldots\ldots\ldots\ldots\ldots\ldots (4)$$

where A is the area of the stream at the section under consideration and Q is the total volume of air moving. But the ratio A_2/A_1 is dependent on the stream angle only, and it is assumed that this angle does not change as the jet velocity is varied. Hence the entrainment ratio at a given distance from the outlet should be independent of the velocity at the outlet face.

Mackey[14] has made a number of analyses based on the momentum theory, with the assumption of uniform velocity across the stream. He has derived equations for the throw and the entrainment of a free stream. His examples deal with both straight-flow and wide-angle streams from rectangular outlets of various aspect ratios. Some numerical results based on these ideal equations are given in Table 2.

Analytical predictions of jet performance such as those given in Table 2, whether based on the momentum theory or otherwise, are all dependent on many simplifying assumptions. Hence they lead to several questions which can be answered only by experiment:

1. Does the total momentum of the stream actually remain constant?
2. Is the ratio of entrained room air to primary air at a given distance from the outlet actually independent of face velocity?
3. Is the throw actually proportional to the face velocity?
4. What are the effects of variations of velocity across the stream, and is there any practical way of taking these effects into account?
5. What are the effects of the type of approach to an outlet?
6. What is the actual angle of spread of the jet from a rectangular outlet, and is this angle affected by the velocity or quantity of air discharged from a given outlet?
7. Can these analyses be applied to wide-spread grilles, ceiling outlets, and outlets of any shape, size or aspect-ratio?

In an effort to answer some of these questions, the following experimental program was undertaken.

Experimental Set-Up and Procedure

The first requirement for checking the momentum theory was to devise some means for taking into account the variations in velocity across an actual air stream. The analytical predictions of Table 2 were computed on the assumption of *uniform* velocity at all points in a plane at right angles to the stream. But when measurements are made across an actual stream, a wide variation in velocities is found. By dividing the stream into small areas and taking velocity measurements within each small area it is possible to remove this chief violation of actual conditions. In this way a much more accurate application of the momentum theory can be made.

Fig. 2 shows the apparatus used for such a stream analysis on the basis of *differential areas*. By means of a grid of 6-in. squares formed by threads stretched across a large frame, the stream was explored with a velometer. A velocity reading was taken in the center of each 6-in. square where the crepe-paper ribbons indicated air movement (see Fig. 2). Since the mass of the air flowing in any small square is wAV, the momentum is:

$$MV = wAV^2 \quad \ldots\ldots\ldots\ldots\ldots\ldots (5)$$

That is, the momentum is equal to the product of the air density, the area, and the velocity squared. These individual momenta were computed for each small square, and the sum of all of them represented an integrated momentum or total momentum for the entire stream. This amounts to a step by step integration based on area increments of 0.25 sq ft.

Velocity measurements were taken over cross-sections of the stream spaced at intervals of 3 ft up to 18 ft from the outlet. The investigation covered several sizes of openings, several types of grilles and types of approach. Tests were made at various face velocities from 400 to

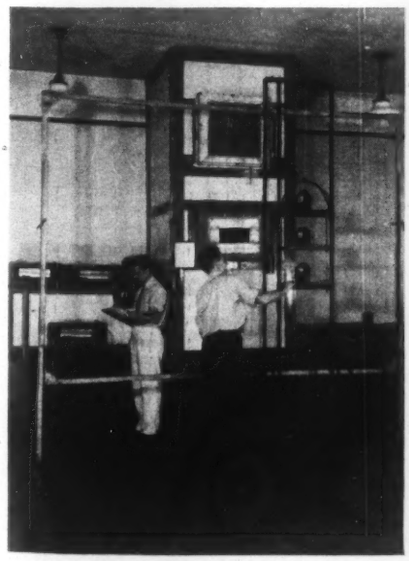


Fig. 2—View of traverse-grid for observing air movement

[14]The Rationale of Air Distribution and Grille Performance, by C. O. Mackey. (*Refrigerating Engineering*, June, 1938, p. 417.)

Table 3—Total Momentum of Air Streams
(First Series)

Outlet Size	Distance from Outlet in Ft	Face Velocity, fpm (Cfm Divided by Gross Area) 400	800	1200	1600
12 in. x 6 in. Free Opening, Duct Approach	0	0 050	0 196	0 458	0 80
	3	0 051	0 200	0 428	0 78
	6	0 053	0 183	0 427	0 81
	9		0 184	0 412	0 71
	12		0 190	0 408	0 74
12 in. x 6 in. Bar Grille, Duct Approach	0		0 1155		0 462
	3		0 0850		
	6		0 1018		0 324
	9		0 0935		0 332
	12		0 0995		0 336
12 in. x 6 in. Nozzle Grille, Duct Approach	0		0 0606		0 2430
	3		0 0401		0 1935
	6		0 0468		0 1562
	9				0 1500
	12				0 1760
12 in. x 6 in. Square-Punched Grille, Duct Approach	0	0 0254	0 1019	0 2300	0 403
	3	0 0273	0 0945	0 1960	0 416
	6	0 0277	0 1039	0 2052	0 379
	9		0 0982	0 2120	0 380
	12		0 0918	0 2235	0 429
	15				0 395
	18				0 404
20 in. x 8 in. Free Opening, Duct Approach	0		0 439	0 998	1 768
	3		0 335	0 944	
	6		0 400	0 825	1 555
	9		0 404		1 600
	12		0 403		
20 in. x 8 in. Bar Grille, Duct Approach	0		0 290	0 530	1 185
	3		0 245		1 058
	6		0 275	0 563	1 060
	9		0 266		0 915
	12		0 260		0 990
14 in. x 4 in. Free Opening, Duct Approach	0			0 327	
	3			0 279	
	6			0 284	
	9				
	12			0 286	
12 in. x 12 in. Free Opening, Duct Approach	0		0 388		
	3				
	6		0 335		
	9		0 321		
	12		0 336		
12 in. x 6 in. Free Opening, Plenum Approach	0	0 1969		0 442	0 786
	3	0 2050		0 520	0 903
	6	0 2305		0 440	0 878
	9	0 2360			0 874
	12			0 525	

Table 4—Total Momentum of Air Streams
(Second Series)

Outlet Size	Distance from Outlet in Ft	Face Velocity, fpm (Cfm Divided by Gross Area) 400	800	1200	2400
8 48 in. x 8 48 in. Free Opening, Plenum Approach	0	0 0750	0 278	0 681	
	9	0 0764	0 276	0 539	
10 285 in. x 7 in. Free Opening, Plenum Approach	0	0 0712	0 282	0 680	
	9	0 0748	0 301	0 585	
12 in. x 6 in. Free Opening, Plenum Approach	0	0 0737	0 299	0 644	1 828
	9	0 0681	0 259	0 557	1 640
14 4 in. x 5 in. Free Opening, Plenum Approach	0	0 0666	0 297	0 648	
	9	0 0651	0 299	0 529	
18 in. x 4 in. Free Opening, Plenum Approach	0	0 0666	0 296	0 648	
	9	0 0625	0 285	0 519	
24 in. x 3 in. Free Opening, Plenum Approach	0	0 0695	0 290	0 648	
	9	0 0657	0 301	0 560	
24 in. x 2 in. Free Opening, Plenum Approach	0		0 187	0 431	
	9		0 215	0 406	
24 in. x 1 in. Free Opening, Plenum Approach	0		0 1040	0 192	0 823
	6		0 1005	0 192	0 668
	9		0 1155	0 202	0 620
24 in. x ½ in. Free Opening, Plenum Approach	0		0 480	0 105	0 416
	6		0 570	0 111	0 351
	9		0 456	0 107	0 344
4 9 in. x 4 9 in. Free Opening, Plenum Approach	0		0 1153	0 212	0 823
	6		0 1030	0 190	0 748
	9		0 1160	0 204	0 642

Table 5—Entrainment Ratios for Free Air Streams

Outlet Size	Distance from Outlet in Ft	Face Velocity, fpm (Cfm Divided by Gross Area) 400	800	1200	1600
12 in. x 6 in. Free Opening, Duct Approach	3	1 77	1 62	1 69	1 67
	6	3 10	2 98	3 17	2 83
	9		4 19	4 51	4 17
	12		5 58	6 45	5 44
12 in. x 6 in. Bar Grille, Duct Approach	3		1 45		
	6		2 76		2 40
	9		4 10		3 82
	12		5 40		5 25
12 in. x 6 in. Nozzle Grille, Duct Approach	1 5		1 17		1 38
	3		1 985		1 985
	6		3 645		3 42
	9				4 60
	12				6 81
12 in. x 6 in. Square Punched Grille, Duct Approach	1 5	1 13	1 03	1 03	
	3	1 92	1 96	1 68	1 94
	6	3 56	3 56	3 52	3 58
	9		5 36	5 08	5 13
	12		6 46	7 39	7 56
	15				8 84
	18				11 2
20 in. x 8 in. Free Opening, Duct Approach	3		1 06	1 16	
	6		2 50	1 85	1 98
	9		3 01		2 94
	12		3 91		
20 in. x 8 in. Bar Grille, Duct Approach	3		1 28		0 93
	6		2 18	1 78	1 93
	9		2 91		2 68
	12		3 82		3 80
14 in. x 4 in. Free Opening, Duct Approach	3			1 58	
	6			3 13	
	9				
	12			5 74	
12 in. x 12 in. Free Opening, Duct Approach	3				
	6		2 00		
	9		2 66		
	12		3 68		
12 in. x 6 in. Free Opening, Plenum Approach	3	1 88		1 78	1 90
	6	3 99		3 54	3 46
	9	5 42			5 40
	12			6 82	

Table 6—Entrainment Ratios for Free Openings
(Plenum Approach)

Outlet Size (Area = 0 5 Sq Ft)	Aspect Ratio	Entrainment Ratio at 9 Ft Face Velocity, fpm (Cfm Divided by Gross Area) 400	800	1200
8 48 in. x 8 48 in.	1 0	4 14	4 36	4 52
10 285 in. x 7 in.	1 47	4 42	4 93	4 70
12 in. x 6 in.	2 0	4 10	4 25	4 45
14 4 in. x 5 in.	2 9	4 01	4 71	4 27
18 in. x 4 in.	4 5	4 66	4 30	4 29
24 in. x 3 in.	8 0	4 27	4 81	4 77

Table 7—Entrainment Ratios for Free-Open Slots
(Plenum Approach)

Size	Aspect Ratio	6 Ft from Outlet: At 200 cfm Volume	At 800 fpm Velocity	At 1200 fpm Velocity	9 Ft from Outlet: At 200 cfm Volume	At 800 fpm Velocity	At 1200 fpm Velocity
24 in. x 3 in.	8				4 27	4 81	4 77
24 in. x 2 in.	12				6 23	6 61	6 27
24 in. x 1 in.	24	6 67	6 81	6 67	8 81	8 12	8 81
24 in. x ½ in.	48	9 20	9 07	10 10	11 98	11 43	12 13

Table 8—Entrainment Ratios for Free Openings
(Plenum Approach)

Size (Area 24 Sq In.)	Aspect Ratio	6 Ft from Outlet: 800 fpm	1200 fpm	2400 fpm	9 Ft from Outlet: 800 fpm	1200 fpm	2400 fpm
4 9 in. x 4 9 in.	1	5 01	4 89	4 72	7 47	7 13	6 88
24 in. x 1 in.	24	6 81	6 67	6 00	8 12	8 81	8 38

2400 fpm. All tests were made in a closed room 48 ft x 21 ft x 13 ft high, the air outlet discharging horizontally from one end of the room, about midway between floor and ceiling. The primary air was measured by the same calibrated orifices used in previous investigations, and the air supply system was also the same.[15,16] The test ducts and plenum chamber were frequently checked for leakage (which was always negligible). In addition to the tests on free openings and grilles in the sizes 12 in. x 6 in., 14 in. x 4 in., and 20 in. x 8 in., tests were also made on square orifices of 4 in. to 12-in. sizes, and on slot-type outlets without grilles, down to one-half inch slot width.

Acknowledgment

The actual tests extended over about six months of steady work, the observers working two at a time. The authors wish to extend special thanks to the following men who were responsible for most of the thousands of observations and calculations here reported: Messrs. R. E. Bolz, Yale University, W. C. Roberts, Lehigh University, D. W. Steel, H. W. Engelman and F. E. Marble, Case School of Applied Science.

Tests were frequently repeated by different observers, to check the accuracy of results. It was found that while the velocities read in each in-

[15] ASHVE Research Report No. 1140—The Use of Air Velocity Meters, by G. L. Tuve, D. K. Wright, Jr., and L. J. Siegel. (ASHVE Transactions, Vol. 45, 1939.) (See Fig. 4.)

[16] ASHVE Research Report No. 1163—Air Flow Measurements at Intake and Discharge Openings and Grilles, by G. L. Tuve and D. K. Wright, Jr. (ASHVE Transactions, Vol. 46, 1940.)

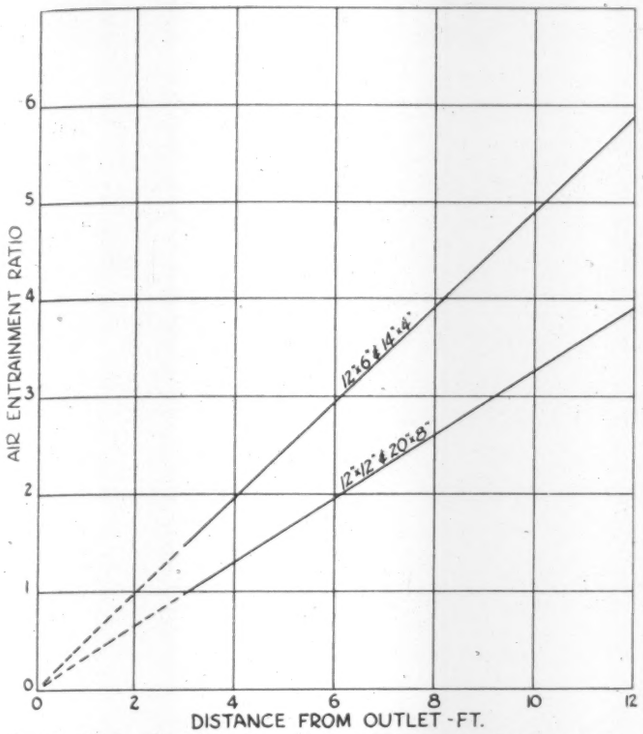


Fig. 3—Effect of outlet area on air entrainment ratio; free openings at end of duct; face velocities 400 to 1600 fpm (for test points see Table 5)

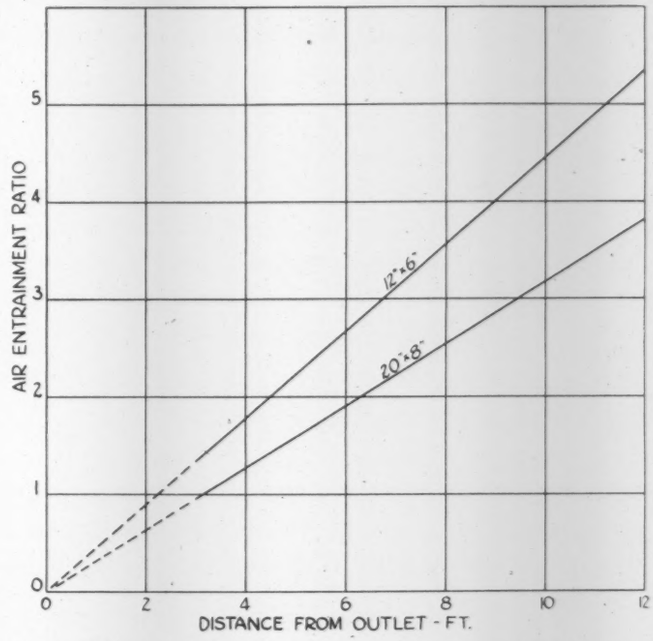


Fig. 4—Effect of grille size on air entrainment ratio; bar grilles, otherwise identical; face velocities 800 to 1600 fpm (for test points see Table 5)

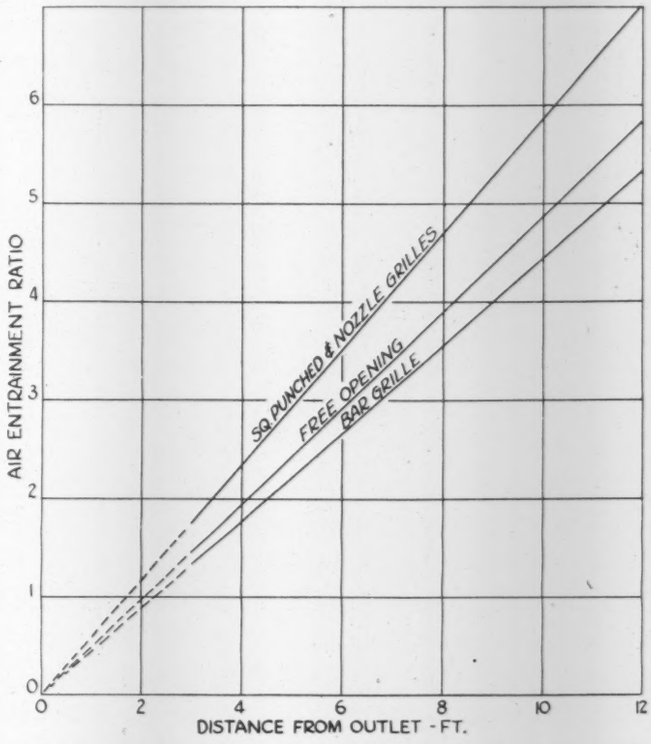


Fig. 5—Effect of type of outlet on air entrainment ratio; 12 in. x 6 in. outlets at end of duct; face velocities 400 to 2000 fpm (for test points see Table 5)

dividual square at different times or by different observers did not agree perfectly, the summation of velocities and summation of momenta over the entire air stream showed very good agreement. As a further precaution several different velometers were used, each velometer being calibrated during the tests. When these calibration corrections were made, the results with different velometers showed satisfactory agreement.

Definitions and Calculations

The test results have been computed in terms of total momentum, jet angle or spread, total throw and total entrainment. *Total volume* at any cross-section of the stream was obtained as the summation of the AV products for all the 6-in. squares and *total momentum* was the summation of the products aAV^2. See Equation (5).

Entrainment ratio is defined as the ratio of the volume of room air entrained to the volume of primary air leaving the outlet. *Total entrainment* is the entrainment ratio measured at the end of the total throw.

Total throw (or blow) is the distance measured perpendicular to the outlet face, at which the velocity has reached a predetermined low value.

Jet angle or spread is the total angle in degrees between the two straight lines on opposite sides of the stream which are drawn from the edge of the outlet and tangent to the edge of the stream. (The spread becomes somewhat less than this as the end of the throw is approached.)

Experimental Data and Results

A tabulation of the total momentum obtained at each test location is summarized in Tables 3 and 4. From these data, the experimental results are seen to substantiate the principle of constant momentum. Since this theory has been substantiated by others also, it is concluded that the accuracy of the method for taking the velocity pat-

tern into account is satisfactory. In general, the mean deviation of the individual momenta from the average of all momenta for the particular test is less than 5 per cent. In view of the unsteady and surging nature of the air stream in a room, this is considered very close agreement. (*Note:* The momentum at zero distance cannot be obtained by the same method as the others; hence it may differ somewhat.)

In terms of the entrainment ratio, or the ratio of entrained room air to primary air, Table 5 shows the results of the same tests that were covered by Table 3. From Table 5 it may be seen that for practical purposes the entrainment ratio may be considered independent of velocity. But the entrainment ratio does vary with the size and shape of the discharge opening, and with the type of grille and the type of approach. A lower entrainment ratio with a larger opening is definitely indicated by the test results of Figs. 3 and 4, although the range of sizes is not great enough to give a definite relation. From these data, the larger opening would also be expected to give a longer total throw with the same outlet velocity, which confirms the results of other investigations and the theoretical calculations of Table 2.

Fig. 5 presents information upon the effect of type of grille upon entrainment ratio by comparing three different styles of grilles and also a free opening, all of 12 in. x 6 in. size. Fig. 6 shows the difference between a straight-duct approach and a plenum approach. This figure may also be taken as a comparison between zero velocity of approach to the outlet, as when the grille is mounted in the wall of a plenum chamber, and maximum approach velocity, as when the grille is mounted at the end of a long duct of the same nominal size. The conclusion might be drawn from Figs. 5 and 6 that streams from square-edged orifice type openings tend to give slightly higher entrainment ratios than streams which are guided by ducts and bars, but the difference is not over 20 per cent.

The effect of the shape of the outlet, expressed as aspect-ratio or as slot width, is indicated by Tables 6 to 8. Table 6 reports tests of six free openings of equal area but of various shapes. These tests show

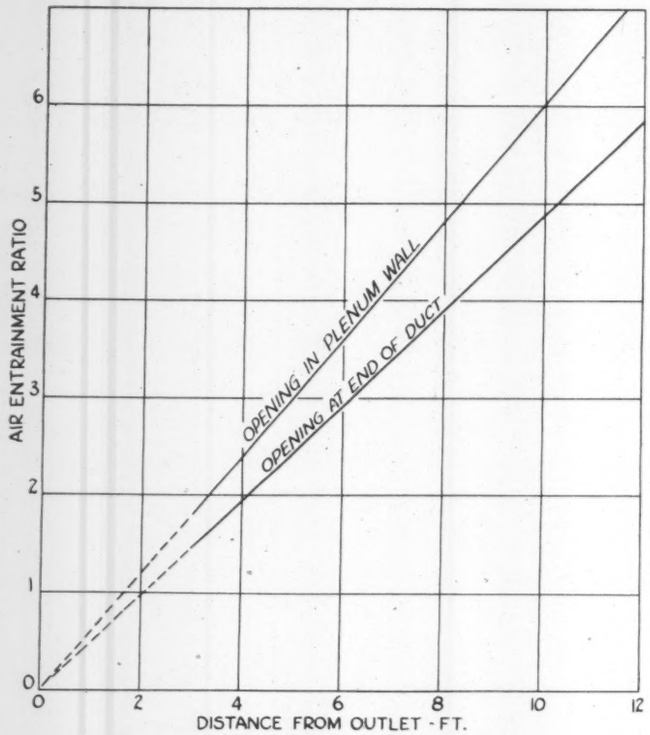


Fig. 6—Effect of type of approach on air entrainment ratio; 12 in. x 6 in. free openings; face velocities 400 to 1600 fpm (for test points see Table 5)

that for ordinary rectangular openings the aspect ratio or shape factor has little effect upon the air entrainment. This agrees with the computed data in Table 2. Table 7 carries the study farther, and gives results on free-open slots from 3 in. wide down to ½ in. wide. Here it is demonstrated that by reducing the slot width to less than 2 in. the entrainment may be markedly increased. Table 8 is a comparison of a 1-in. slot and a square outlet, both of the same total area. The effect of changing the aspect ratio is not as great as might be expected.

Limitations of the testing space available made it difficult to measure the total throw. Table 9 gives the results of a few tests in which the face velocity was varied and the total throw was measured. In these tests the end of the throw was defined as the plane at which the maximum velocity measured by the velometer was 100 fpm.

This relation between average velocity and maximum velocity is of much practical importance, especially when measuring total throw or in specifying conditions which shall be met in order to eliminate *drafts*. The results of extensive studies show that the maximum velocity is usually from 2.5 : 3.5 times the average velocity across the section as measured by the traverse-grid method.

Typical data sheets from which these and the other results were

8.48" × 8.48" Free Opening Vel. = CFM / Area = 1200
Test No. 20 Date 8-13-41 Distance - 9 ft.
Observers: G.B.P. & H.W.E.

			0	0	0	0	0	0			
		0	90	90	130	80	50	20	0		
	0	10	170	170	190	200	210	60	0	0	
	0	150	280	300	300	290	300	160	40	50	0
	0	160	210	290	320	300	290	240	110	60	0
0	10	120	180	310	410	440	300	300	130	50	0
	0	110	150	300	400	360	290	270	160	70	0
	0	50	100	300	300	300	300	210	120	20	0
		0	50	200	150	180	200	130	60	0	
			0	10	50	90	70	20	10	0	
				0	0	10	0	10	0		
					0	0	0	0			

Fig. 7—Typical data sheet showing maximum velocity of air as measured by traverse-grid method

Table 9—Distances from Outlet in Feet Where Maximum Velocity was 100 fpm

Outlet Size	Area Sq Ft	Aspect Ratio	Velocity = Cfm Divided by Gross Area				
			400 fpm	600 fpm	800 fpm	1200 fpm	2400 fpm
8.48 in. x 8.48 in.	0.5	1.0	18	.....	32	.....	.....
10.285 in. x 7 in.	0.5	1.47	18	.....	36	.....	.....
12 in. x 6 in.	0.5	2.0	18	.....	36	.....	.....
14.4 in. x 5 in.	0.5	2.9	18	.....	35	.....	.....
18 in. x 4 in.	0.5	4.5	17	.....	32	.....	.....
24 in. x 3 in.	0.5	8.0	16	.....	32	.....	.....
4.9 in. x 4.9 in.	0.167	1	.....	17	24	35	.....
24 in. x 1 in.	0.167	24	.....	.....	18	27	.....
24 in. x ½ in.	0.083	48	.....	.....	12	18	36

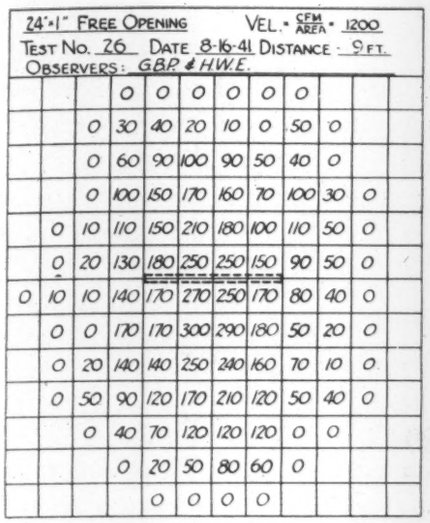


Fig. 8—Typical data sheet showing maximum velocity of air as measured by traverse-grid method

obtained are shown in Figs. 7 and 8. These sheets give the uncorrected velometer readings as observed at the grid, and the outlet size is superimposed upon each traverse area, in proper position and to scale.

Jet angles were measured in all tests, and Table 10 gives a summary of some of the results. Angles were determined experimentally from layouts to scale, using data sheets, similar to Figs. 7 and 8 for distances of 3 to 12 ft from the outlet. The *average* jet angle, which is taken as the arithmetical mean between the horizontal and the vertical angle of spread, tends to increase slightly with increasing outlet velocity. For a slot-type outlet the horizontal spread is not the same as the vertical spread, and the stream tends to become circular. A few tests were made on directional-flow grilles with wide-spreading stream, and these indicated that very much greater air entrainment may be obtained with this type of outlet, as predicted by the results in Table 2.

Such items as directional flow grilles and heating and cooling of the air stream, should be covered by further investigation. These studies are to be continued under the direction of the Technical Advisory Committee on Air Distribution and Air Friction of the ASHVE, as already authorized by the Research Committee of the Society. Critical comment and suggestions for this work will be gladly received.

Conclusions from Tests

The following conclusions may be drawn as a summary of the experimental results here presented. It will be noted that the tests were limited to air outlets less than 160 sq in. in area and outlet velocities up to 2400 fpm, but that a variety of types and sizes were tested.

1. The total momentum of a free air stream remains practically constant throughout its measurable length.
2. Entrainment of room air is continuous throughout the length of an air stream, and amounts to 20 to 70 per cent of the primary air volume per foot of travel or throw. This percentage entrainment may be increased greatly by using slot-type outlets or wide-angle spreading grilles.
3. The ratio of entrained air to primary air *at a given distance* from the outlet face varies as follows:
   (a) Entrainment ratio decreases as the area of the outlet is increased.
   (b) Entrainment ratio is practically independent of the outlet face velocity.
   (c) Entrainment ratio is practically the same for all square and rectangular outlets of the same area, except the very narrow slots.
4. The total throw or blow of an air stream is directly proportional to the outlet velocity.
5. The air stream from a narrow-slot outlet becomes almost circular toward the end of its throw.
6. The maximum velocity at any plane at right angles to the stream is usually about three times the average velocity in that plane.
7. The *average* jet angle for the stream from a rectangular outlet of any shape, without spreading vanes, is about 19 deg, ±5 deg depending on type of approach, type of outlet and velocity.
8. The air entrainment by a venturi-mixer unit is small compared with the natural entrainment by a free air stream.
9. For increasing the air mixing and reducing the throw, the advantages of the narrow slot or of the narrow-ring ceiling outlet are clearly indicated, as are those of the wide-spreading grille.

Table 10—Jet Angles in Degrees

Free Openings, Plenum Approach	Face Velocity, fpm (Cfm Divided by Gross Area)								
	400			800			1200		
	Hor	Vert	Av	Hor	Vert	Av	Hor	Vert	Av
8.48 in.² x 8.48 in.	21	20	20.5	23	25	24	24	28	26
10.285 in. x 7 in.	20	20	20	22	26	24	23	28	25.5
12 in. x 6 in.	17	23	20	18	26	22	20	28	24
14.4 in. x 5 in.	16	24	20	17	29	23	21	30	24.5
18 in. x 4 in.	14	26	20	14	28	21	16	31	23.5
24 in. x 3 in.	10	31	20	14	30	22	15	34	24.5
4.9 in. x 4.9 in.	20	22	21	20	22	21	22	21	21.5
24 in. x 1 in.	10	26	18	10	28	19	14	32	23
24 in. x ½ in.	10	30	20	11	31	21	12	32	22

STANDARD SAFETY SIGNS

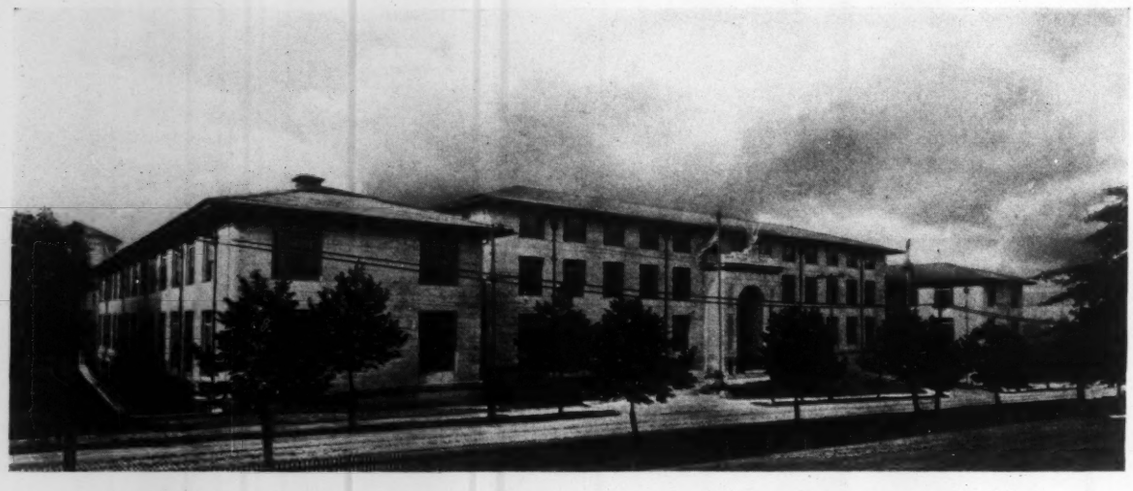
A new standard for industrial accident prevention signs which provides for uniform colors and wording has been announced by the *American Standards Association*. The purpose is to increase the effectiveness of safety signs in preserving human life.

For maximum safety, it should not be necessary to stop, read, and analyze the meaning of each particular warning sign. Reaction to such signs should be automatic, particularly in workplaces where employees cannot read English readily but can be trained to recognize instantly and be warned by standard sign designs and colors.

It is with this thought in mind that the committee developing the standard has divided all accident prevention signs into five major groups:

1. Danger signs to be used to warn about specific dangers only and removed as soon as the danger they warn against no longer exists.
2. Caution signs used to warn against possible danger or unsafe practices.
3. Safety instruction signs providing information regarding general safe practices.
4. Directional signs used to point the way to staircases, fire escapes, exits, etc.
5. Informational signs used to carry messages of a general nature such as rules and regulations.

American Standard Specifications for Industrial Accident Prevention Signs—Z35.1-1941 has been published in pamphlet form.



Pittsburgh Experiment Station of the U. S. Bureau of Mines where the Research Laboratory of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS is located

A Review of Glass Transmittance Coefficients

By F. C. Houghten*, D. Sc., M. L. Carr**, and David Shore***, Pittsburgh, Pa.

DURING the past quarter of a century several investigators have conducted research on heat transmission through glass. This research has included in its scope not only the determination of the heat transmittance coefficients of glass alone, but also of glass as used in various window arrangements. That is, the tests have dealt with plain glass, and also with single and multiple glazed windows set in both wood and metal sash. This paper, intended as a review of the more pertinent data gleaned from these investigations, includes a resumé of the methods employed and the results obtained in connection with the investigations, and further, contains an attempt at some comparison and correlation of the data gathered.

One of the earlier investigators in this particular field of research was John R. Allen, who conducted his work on heat losses through glass in 1916.[1] His apparatus consisted of a rectangular wooden box, having thick, well-insulated walls, with one of its sides removable to facilitate the insertion of the panel to be tested. Moreover, the box was placed on trunnions so that it could be turned in any position, thus enabling the panel to be tested either as a floor, wall, or ceiling.

The box was placed in an outdoor shelter, and in order to simulate ordinary outdoor atmospheric conditions for the outside of the box, a blower served to create wind velocity effects and spray nozzles produced rain effects. The inside was maintained at a uniform temperature by electric heaters consisting of ordinary carbon filament lamps. These lamps were screened by double shields of brown paper in order to reduce the radiation of heat to the test panel and to the eight thermocouples placed in different parts of the box to measure the inside temperature. Outside air temperature was determined by six mercury thermometers. A recording watt-hour meter was used to measure the electrical energy supplied to the box.

Each test was conducted with all conditions being as nearly constant as possible, particular care being given to the maintenance of a uniform temperature difference between the inside and outside air. The length of a test was determined by the outside temperature, it being thought desirable to have a period of at least four hours of constant temperature before starting and also at the end of the test. The loss in heat through the walls of the box itself was determined by placing in the open side of the box a section constructed in the same manner as the rest of the box. The coefficient thus determined was then applied to all the box surfaces exposed to the outside during the test. The results obtained from Allen's tests on glass, as well as those for other experimenters, are listed in Table 1.

Also in 1916, Hammond and Holmberg[2] conducted heat transmission tests on glass at the Thermal Testing Plant of Pennsylvania State College. The experiments were made with a cubical corkboard box suspended in a large calorimeter room which was cooled by brine coils. A fan and an electric heating coil were placed inside the box for producing the required, uniformly distributed temperature difference between the inside and the outside of the box. The electrical energy

*Director, Research Laboratory, ASHVE, Pittsburgh, Pa. MEMBER of ASHVE.
**Director, Pittsburgh Testing Laboratory, Pittsburgh, Pa. MEMBER of ASHVE.
***Research Engineer, ASHVE Research Laboratory, Pittsburgh, Pa. JUNIOR MEMBER of ASHVE.

[1]Heat Transmission Through Building Material, by John R. Allen. (ASHVE TRANSACTIONS, Vol. 22, 1916, p. 507.)

[2]A Study of Surface Resistance with Glass as a Transmission Medium, by H. R. Hammond and C. W. Holmberg. (*ASME Transactions*, Vol. 39, 1917, p. 757.)

input into the box was measured by accurately calibrated voltmeters and ammeters on both the heating coil and fan circuits, and the temperatures were recorded by platinum-resistance thermometers. These resistance thermometers were carefully calibrated and then protected from any air currents that might affect the readings. The corkboard box was calibrated for heat loss and then the removable side was replaced by a four light window, the plate glass being ⅛ in. thick. Thermometers were placed at distances of ¼ in., ½ in., 1 in., 3 in., 10 in., and 28 in. from the outside surface, while the inside air temperature was taken at a point 10 in. from the inside surface of the glass. Readings were taken until constant temperatures had been maintained for some time.

Some experiments to determine the rate of heat transmission through single and double glazing set in various types of sash were conducted in 1916 by W. S. Brown, under the direction of A. N. Sheldon.[3] The sash tested were of such size as to completely fill an opening in the face of a test box constructed of 2-in. tongued and grooved white pine, well-seasoned and covered with 2-in. cork boards, the joints of the latter being made air tight with brine cement. To reduce to a minimum the leakage around the sash at its line of contact with the box, a continuous wool felt gasket, 1 in. thick, was placed around the edge, and the sash screwed firmly against it. Heat was supplied by an electric heater placed at the bottom of the test box, and in order to reduce direct radiation of heat through the lower panes of glass, ¼ in. white asbestos-board plates were fastened against the framework of the heater.

A fair average of the internal air temperature of the box was obtained from the readings of 12 thermometers. The temperature of the outside air, heated by a steam radiator, was controlled by a thermostat and observed by an accurate wall thermometer hung halfway between two air circulating fans. These fans directed air against the sash at an angle of 30 deg, thus providing a positive circulation over it of approximately 7 mph as read from an anemometer. The back and sides of the test box were protected against air currents by vertical baffle boards placed on these sides.

In order to determine what portion of the total heat measured was conducted through the sash in a given test, it was necessary to subtract from the total quantity the amount lost through the back, sides, top and bottom surfaces of the box. This latter amount was ascertained by means of *blank-run* tests, the front of the box consisting of the same materials as the other sides. This procedure was quite similar to that of Allen's mentioned previously. During the *blank-run* tests, the fans were not in operation, as the purpose of these tests was to obtain the rate of transmission through the walls of the box under the conditions of the main tests; that is, with still air for the back, sides, top and bottom. Furthermore, the error due to a possible change in conductivity of the box itself during experiments was determined by tests made at temperature differences of 40 and 70 F, both before and after the sash tests. It was found that a slight increase in conductivity occurred, probably as a result of shrinkage and warping of the boards.

During all tests temperatures of the different types of sash were measured in order to ascertain, to some degree, the effect of these sash on the rate of heat transmission. Thermometers placed tightly against the sash surfaces were used to make

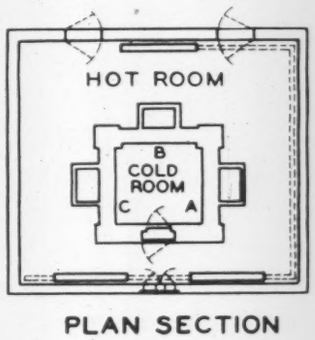


Fig. 1*a*—Plan section of the laboratory building and the hot room used by Kreuger and Eriksson

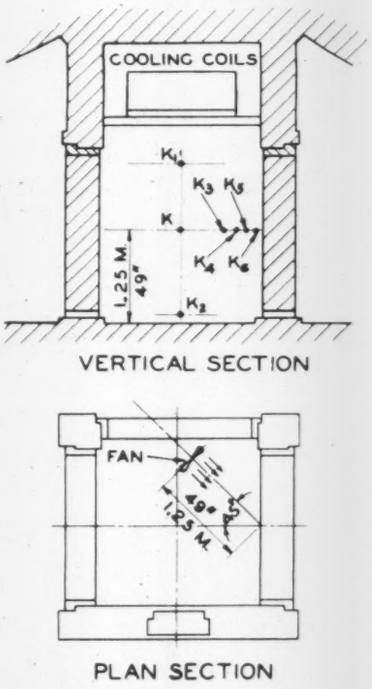


Fig. 1*b*—Sectional views of the cold room used by Kreuger and Eriksson

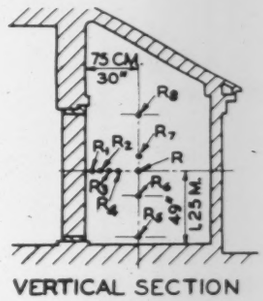


Fig. 1*c*—Location of thermometers used in Kreuger and Eriksson's hot room on warm side of test wall

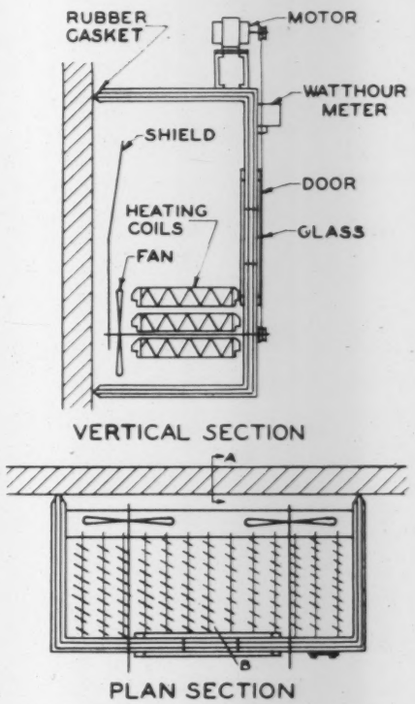


Fig. 1*d*—Sectional views of the hot box used by Kreuger and Eriksson

[3]Heat Transmission Through Various Types of Sash, by A. N. Sheldon. (*ASME Transactions*, Vol. 38, 1916, p. 857.)

these measurements, and although the temperatures recorded were probably slightly less than actual, it was believed that adequate comparison was furnished by these readings.

During the period 1922 to 1924 a comprehensive study of heat transmission through building materials, including glass, was carried on by H. Kreuger and A. Eriksson[4] of the Academy of Engineering Science at Stockholm, Sweden. A square room situated in the approximate center of a large, heated, and well-insulated space was used for the various tests. One wall of this room was fixed in position while the other three were made removable and replaceable by the three panels of the particular construction to be tested. In order to provide natural circulation, cooling was accomplished by refrigeration coils placed at the top of the room. The removable walls of the cold room are shown in Fig. 1*a* at A, B, and C, respectively.

Each of the hot boxes, as shown in Figs. 1*a* and 1*d*, was placed with its open side against the center of one of the three wall panels to be tested, air-tight contact being obtained by means of a rubber gasket. The walls of the hot box consisted of two layers of wood with insulating felt between to minimize the heat loss and to facilitate temperature control. The heating coils were placed near the bottom of the box with two fans directly in front of them. The motor driving the fans was placed outside the box so as to minimize the effect on the temperature inside the box. For protection against direct radiation a shield was placed between the heaters and the test wall. The air inside the hot box was kept at the same temperature as that of the surrounding room so as to eliminate heat flow to or from the hot box except that through the test wall, and this amount of heat was readily obtained from voltmeter-ammeter readings. While fan circulation was used for the inside of the hot box, only natural air circulation was used for the cold room wall surface exposed outside the hot box. The fans were then run at a speed to give a temperature distribution at the wall surface inside the hot box the same as that at the surface outside the box.

Fan circulation was also provided in the cold room in order to maintain as nearly uniform temperature distribution as possible. Fig. 1*b* gives two sectional views of the cold room and shows the points at which temperature observations were made. Thermometer K_6 was placed in contact with the wall surface, and the other thermometers at varying distances from the surface. Experiments showed that the mean temperature occurred at a distance of 49 in. from the floor, at which elevation all thermometers except K_1 and K_2 were placed. The circulating fan is shown in the plan section located 49 in. from both the floor and test wall, delivering air at an angle of 45 deg.

The location of the five thermometers placed in a vertical plane in the hot room is seen in Fig. 1*c*. As in the case of the cold room the mean temperature occurred at a height of 49 in. above the floor. Therefore, four thermometers were placed at this elevation above the floor to give the horizontal temperature variation. In cases where the temperatures of the test wall surfaces were measured with thermometers, a type of instrument with a flattened bulb was used. The bulb was placed in a shallow depression in the wall surface, the depth of which was half the thickness of the bulb, and the exposed surface of the bulb was then covered with plaster of Paris. The bulbs of the thermometers used for measuring air temperatures were surrounded with paper shields to avoid radiation effects; it was found, however, that the shields made practically no difference in the readings of the thermometers. Thermocouples were used to measure temperatures inside the hot box.

After the test walls were placed in position as sides of the cold room, the room was then cooled to about zero degrees Fahrenheit. The hot box, thermometers, and thermocouples were then placed in position, and the electrical heaters were put into operation for 24 hours or more until conditions became constant. Once established, all temperatures were kept as nearly constant as possible, and complete observations made hourly over a period of at least 8 hours.

In 1926 F. B. Rowley,[5] at the University of Minnesota, conducted a series of experiments employing the hot box method of testing to determine the best spacing for double glazed windows. Fig. 2 shows a sectional view of the hot box and the wall section in place, indicating the location of the 8 heating elements used in the inner box and a like number in the outer box. The heating elements were distributed with the greatest number near the bottom, the location of the elements being determined by experiment in order to get as nearly uniform temperature distribution as possible throughout the two boxes. In order to maintain uniform temperatures throughout the boxes two 8-in. fans were installed in each box, as shown; and further, thermostatic control was used to maintain the hot box temperature. The fans for the inner box were driven by an

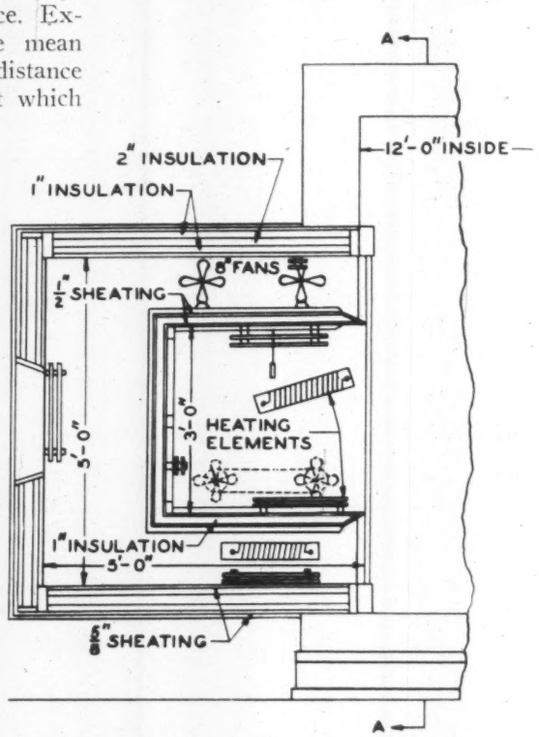


Fig. 2—Sectional view of hot box used by Rowley

[4]Investigations of Heat Transmission Through Various Building Constructions, by H. Kreuger and A. Eriksson. (Report of the *Royal Academy of Scientific Research*—Theses No. 7 and 36, Stockholm, Sweden.)

[5]Some Results of Heat Transmission Research, by F. B. Rowley. (ASHVE Transactions, Vol. 32, 1926, p. 199.)

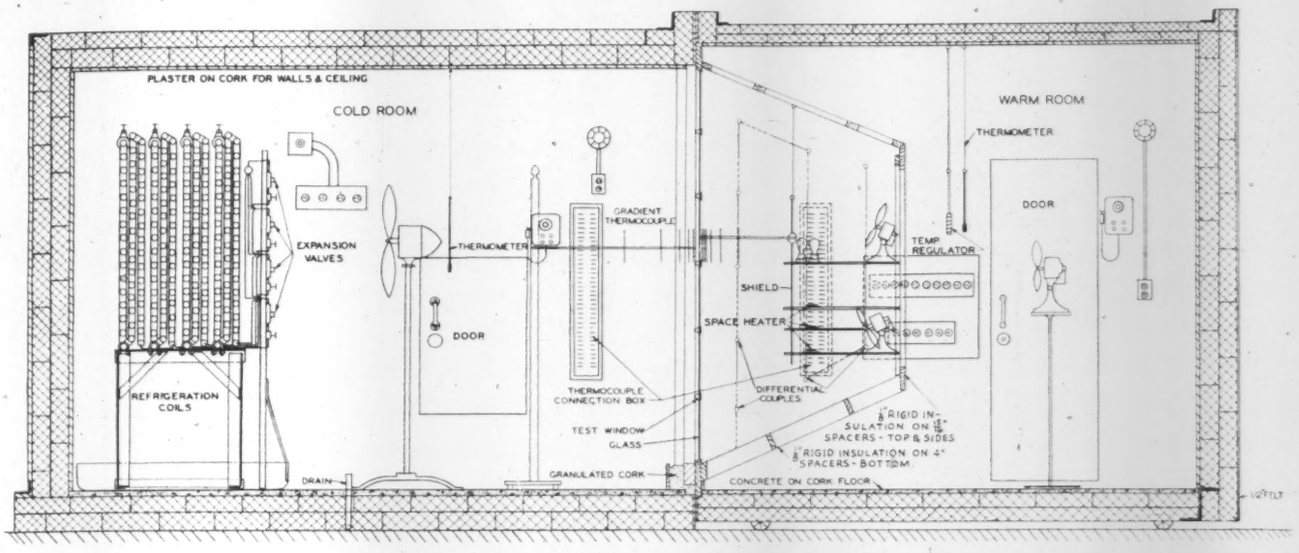


Fig. 3—Cross section of apparatus used by Carr

externally mounted motor and run slow enough to maintain uniform temperature over the test surface and yet have practically still air conditions.

The air temperature on the cold side of the test specimen was maintained by an ammonia compression refrigerating machine by direct expansion into piping in the cold room. However, it was difficult to obtain sufficiently accurate temperature control by use of the expansion valves alone; therefore, a small thermostatically controlled heating unit was installed. The refrigerating machine was set to give temperatures slightly lower than required and the heating unit was set for the final control.

All temperatures were measured by copper-constantan thermocouples and a potentiometer. During the period that the test specimen was being brought to constant temperature conditions a recording potentiometer was used, but after the test was started all readings were taken with a precision potentiometer. The arrangement of the thermocouples was as follows: One group of five, centrally located 2 in. from each of the inside surfaces excepting the test panel surface, was used to measure the air temperature of the inner box. A second group of five couples, each centrally located 2 in. from the outer surface of the sides of the inner box, was used to measure the air temperature in the outer boxes. These two groups of couples were then used in balancing the temperatures of the inner and outer hot boxes. To determine the air temperatures on the hot and cold sides of the test wall, a group of five couples was placed in the inner hot box 6 in. from the test surface, and another group of four thermocouples was placed in the cold room also 6 in. opposite the test surface. The thermocouples were so arranged that their temperatures could be read separately or in groups; that is, the average temperatures in the inner hot box, the average temperatures in the outer hot box, or the average temperatures of the air on either side of the test section could be obtained from a single reading. Other thermocouples were used to determine surface temperatures of the test specimen or any other temperatures required.

A paper[6] was prepared from data collected in 1932 by the Pittsburgh Testing Laboratory on the rate of heat transmission through single- and double-glazed windows. The hot box method of testing was employed in this work also, a cross-section of the apparatus being shown in Fig. 3. The apparatus consisted of a cold room in which outdoor air conditions were simulated, and a warm room which contained the guarded enclosure where the indoor air conditions were reproduced. The test specimen served as the dividing partition between these two rooms.

The walls, floor, and ceiling of the cold room were constructed of 8-in. cork plastered on the inside, and a cork insulated vestibule with refrigerator doors provided access when the open end of the room was closed by the test panel. The cooling system consisted of four independent compressor units, each connected to its own cooling coil located in the cold room, the total capacity of the system being sufficient to maintain a cold room temperature of —30 F. The warm room made of similar construction was mounted on casters to permit its removal from the cold room during the installation of a test specimen. A flexible system of electrical heating was controlled either automatically by a mercury thermostat or manually, as desired. Both the warm and cold rooms were provided with fans for air circulation, electric lights, electric outlets, and connections for thermocouples and telephones. The guarded enclosure, shaped like the frustum of a four-sided pyramid, had its large open end of the same size as the windows to be tested, and was fastened with a calked air-tight joint to the wood frame in which the test specimen was installed. The walls were of double insulated construction, and contained about 50 differential couples in series threaded through the walls, with approximately one couple in each 2 sq ft of the inside and outside wall surfaces. The guarded enclosure was provided with its own set of heaters, shielded from the test panel, fans and lights, and the electric input to all these appliances was measured.

[6]Heat Transfer Through Single and Double Glazing, by M. L. Carr, R. A. Miller, Leighton Orr and Alan C. Byers. (ASHVE TRANSACTIONS, Vol. 44, 1938, p. 471.)

Table 1—Transmittance Coefficients for Flat Glass and Windows

Ref. No.	Authority [a]	Trans. Coeff. *U* Btu/Hr/Sq Ft/Deg F	Glass Set Up: No. Sheets Glass	Air Space	Area: Total	Area: Test	Description	Hot Side: Temp.	Hot Side: Vel.	Cold Side: Temp.	Cold Side: Vel.	Mean Temp. F
1	A	0.64	1	—	16 Sq Ft		4 Lights Metal Sash		Still Air		Still Air	
2	H & H	1.10	1	—	25 Sq Ft		4 Lights Wood Sash		Still Air		Still Air	40.0
3	S	1.19	1	—	4 Ft 2¾ In. x 7 Ft 9¼ In.		20 Lights 12″ x 18″—¼″ Rough Wire Glass—Solid Rolled Steel Sash		Still Air		7 Mph	95.0
4		1.06	1	—	4 Ft 2¾ In. x 7 Ft 9¼ In.		Same as (3) except for Wood Sash		Still Air		7 Mph	95.0
5		1.10	1	—	4 Ft 2¾ In. x 7 Ft 9¼ In.		Same as (3) except for Hollow Metal Sash		Still Air		7 Mph	95.0
6		0.85	2	¾″	4 Ft 2¾ In. x 7 Ft 9¼ In.		20 Lights 12″x18″—Outer Glass ¼″ Wire — Inner Glass ⅛″ Plate—Solid Steel Sash		Still Air		7 Mph	95.0
7		0.94	2	3/16″	4 Ft 2¾ In. x 7 Ft 9¼ In.		20 Lights 12″x18″—Outer Glass ⅛″ Rib—Inner Glass ⅛″ Plate—Solid Steel Sash		Still Air		7 Mph	95.0
8		0.58	2	5/8″	4 Ft 2¾ In. x 7 Ft 9¼ In.		20 Lights 12″x18″—Outer Glass ¼″ Wire—Inner Glass D. S. Plate—Wood Sash		Still Air		7 Mph	95.0
9		0.63	2	¾″	4 Ft 2¾ In. x 7 Ft. 9¼ In.		Same as (6) except for Hollow Metal Sash		Still Air		7 Mph	95.0
10	K & E	1.02	1	—	1.2 M² (12.9 Ft²)	1.0 M² (10.8 Ft²)	Plain Glass	65.7	Two Fans Operated at a Speed Designed to Give a Temp. Distribution at the Wall Surface Inside the Box Equal to that at the Wall Surface Outside the Box.	—3.1	Nat. Circ.	31.3
11		0.63	2	None	1.2 M² (12.9 Ft²)	1.0 M² (10.8 Ft²)	Plain Glass	68.0		—0.4	Nat. Circ.	33.8
12		0.53	2	1 Cm. (0.394″)	1.2 M² (12.9 Ft²)	1.0 M² (10.8 Ft²)	Plain Glass	68.0		—3.5	Nat. Circ.	32.3
13		0.49	2	1 Cm. (0.394″)	1.2 M² (12.9 Ft²)	1.0 M² (10.8 Ft²)	As above with 1M² Spacing Frame Between the Glass Lights	69.0		—4.7	Nat. Circ.	32.2
14		0.51	2	1 Cm. (0.394″)	1.2 M² (12.9 Ft²)	1.0 M² (10.8 Ft²)	Same as (13) except with 4 Vertical Dividing Rods	68.9		—4.9	Nat. Circ.	32.0
15		0.51	2	1 Cm. (0.394″)	1.2 M² (12.9 Ft²)	1.0 M² (10.8 Ft²)	Same as (13) except with 4 Horizontal Dividing Rods	68.4		—5.8	Nat. Circ.	31.3
16		0.47	2	10 Cm. (3.94″)	1.2 M² (12.9 Ft²)	1.0 M² (10.8 Ft²)	Plain Glass	64.4		1.4	Nat. Circ.	32.9
17		0.45	2	10 Cm. (3.94″)	1.2 M² (12.9 Ft²)	1.0 M² (10.8 Ft²)	Same as (16) with 1M² Spacing Frame Between the Glass Lights	68.0		—2.7	Nat. Circ.	32.9
18		0.45	2	10 Cm. (3.94″)	1.2 M² (12.9 Ft²)	1.0 M² (10.8 Ft²)	Same as (17) with 4 Vertical Dividing Rods	64.8		—0.8	Nat. Circ.	32.0
19		0.45	2	10 Cm. (3.94″)	1.2 M² (12.9 Ft²)	1.0 M² (10.8 Ft²)	Same as (17) with 4 Horizontal Dividing Rods	68.0		1.8	Nat. Circ.	34.9
20		0.28	3	5 Cm. (1.97″)	1.2 M² (12.9 Ft²)	1.0 M² (10.8 Ft²)	Plain Glass	68.4		—2.9	Nat. Circ.	32.7
21		0.22	4	3.3 Cm. (1.31″)	1.2 M² (12.9 Ft²)	1.0 M² (10.8 Ft²)	Plain Glass	67.8		—7.8	Nat. Circ.	30.0
22		0.77	1	—	1.3 M² (14.0 Ft²)	1.0 M² (10.8 Ft²)	9 Lights 40 Cm. x 40 Cm.—Wood Sash (15¾″ x 15¾″)	66.2		—0.9	Nat. Circ.	32.5
23		1.26	1	—	1.3 M² (14.0 Ft²)	1.0 M² (10.8 Ft²)	9 Lights 40 Cm. x 40 Cm.—Wood Sash (15¾″ x 15¾″)	62.4		6.8	5.6 Mph	34.6
24		0.57	2	2Mm (0.079″)	1.3 M² (14.0 Ft²)	1.0 M² (10.8 Ft²)	9 Lights 40 Cm. x 40 Cm.—Wood Sash (15¾″ x 15¾″)	66.7		—4.7	Nat. Circ.	31.0
25		0.83	2	2 Mm (0.079″)	1.3 M² (14.0 Ft²)	1.0 M² (10.8 Ft²)	9 Lights 40 Cm. x 40 Cm.—Wood Sash (15¾″ x 15¾″)	67.1		—3.3	5.6 Mph	31.9
26		0.59	2	2 Mm (0.079″)	1.3 M² (14.0 Ft²)	1.0 M² (10.8 Ft²)	9 Lights 42 Cm. x 42 Cm.—Iron Sash (16½″ x 16½″)	68.0		—2.7	Nat. Circ.	32.7
27		0.47	3	2 Mm (0.079″)	1.3 M² (14.0 Ft²)	1.0 M² (10.8 Ft²)	Same as (22)	67.5		—3.1	Nat. Circ.	32.2
28		0.63	3	2 Mm (0.079″)	1.3 M² (14.0 Ft²)	1.0 M² (10.8 Ft²)	Same as (22)	67.6		—2.9	5.6 Mph	32.3
29		0.42	2	3.5 Cm. (1.38″)	1.3 M² (14.0 Ft²)	1.0 M² (10.8 Ft²)	Same as (22)	65.8		—0.9	Nat. Circ.	32.5
30		0.39	2	10 Cm. (3.94″)	1.3 M² (14.0 Ft²)	1.0 M² (10.8 Ft²)	Same as (22)	67.3		—2.9	Nat. Circ.	32.2
31		0.51	2	10 Cm. (3.94″)	1.3 M² (14.0 Ft²)	1.0 M² (10.8 Ft²)	Same as (22)	66.2		—2.7	Nat. Circ.	31.8
32	R	0.89	1	—	25 Sq Ft	9 Sq Ft	Plain Glass	80.1	Fans Operated at Speed to Give Natural Air Circulation	4.0	Fans Operated at Speed to Give Natural Air Circulation	42.1
33		0.71	2	⅛″	25 Sq Ft	9 Sq Ft	Plain Glass	77.5		—10.3		43.9
34		0.59	2	¼″	25 Sq Ft	9 Sq Ft	Plain Glass	80.6		—11.5		46.0
35		0.57	2	⅜″	25 Sq Ft	9 Sq Ft	Plain Glass	80.1		—15.2		47.6
36		0.50	2	½″	25 Sq Ft	9 Sq Ft	Plain Glass	80.3		—14.3		47.3
37		0.49	2	⅝″	25 Sq Ft	9 Sq Ft	Plain Glass	78.9		—13.7		46.3
38		0.47	2	¾″	25 Sq Ft	9 Sq Ft	Plain Glass	79.5		—14.3		46.9
39		0.49	2	1″	25 Sq Ft	9 Sq Ft	Plain Glass	80.5		—1.3		40.9
40		0.49	2	1¼″	25 Sq Ft	9 Sq Ft	Plain Glass	79.8		—10.8		45.3
41		0.49	2	2″	25 Sq Ft	9 Sq Ft	Plain Glass	81.5		—0.2		40.8
42		0.46	2	2½″	25 Sq Ft	9 Sq Ft	Plain Glass	80.5		—0.2		40.4

[a]—See footnote on next page.

Table 1 (Continued)—Transmittance Coefficients for Flat Glass and Windows

Ref. No.	Authority	Trans. Coeff. *U* Btu/Hr/Sq Ft/Deg F	Glass Set Up: No. Sheets Glass	Glass Set Up: Air Space	Glass Set Up: Area Total	Glass Set Up: Area Test	Glass Set Up: Description	Hot Side: Temp	Hot Side: Vel.	Cold Side: Temp.	Cold Side: Vel.	Mean Temp. F
43	P.T.L.	1.10	1	—	7 Ft 4⅜ In. Sq		36 Lights Steel Sash	70.0	2.18 Mph	0	0.66 Mph	35.0
44		0.99	1	—	7 Ft 4⅜ In. Sq		Same as (43) except for 9 Lights	70.0	2.18 Mph	0	0.66 Mph	35.0
45		0.71	2	¼″	7 Ft 4⅜ In. Sq		36 Lights Steel Sash	70.0	2.18 Mph	0	0.66 Mph	35.0
46		0.60	2	¼″	7 Ft 4⅜ In. Sq		Same as (45) except for 9 Lights	70.0	2.18 Mph	0	0.66 Mph	35.0
47		0.65	2	⅛″	7 Ft 4⅜ In. Sq		Same as (45) except for 9 Lights	70.4	2.18 Mph	—29.0	0.66 Mph	20.7
48		0.56	2	⅜″	7 Ft 4⅜ In. Sq		Same as (45) except for 9 Lights	71.6	2.18 Mph	—31.0	0.66 Mph	20.3
49		0.55	2	½″	7 Ft 4⅜ In. Sq		Same as (45) except for 9 Lights	70.2	2.18 Mph	—0.9	0.66 Mph	34.6
50		0.54	2	⅝″	7 Ft 4⅜ In. Sq		Same as (45) except for 9 Lights	70.2	2.18 Mph	—0.1	0.66 Mph	35.0

A—Allen
H & H—Hammond & Holmberg
S—Sheldon
K & E—Kreuger & Eriksson
R—Rowley
P.T.L.—Pittsburgh Testing Laboratory

Temperature measuring apparatus consisted of 48 copper-constantan thermocouples, 24 in the cold room and 24 in the warm room and guarded enclosure. Terminal boxes for the thermocouples were permanently installed in the two rooms, and cables led from these boxes to selector switches on an instrument bench. The electrical input was measured by an integrating watt-hour meter, an indicating wattmeter serving as a check. Light threads were used to determine the location and direction of air movement, while a Kata thermometer was used to measure the velocities produced by the fans. Humidity, which was partially controlled by silica gel, was measured by a mercury thermometer type of hygrometer and by a set of wet and dry junction thermocouples placed in an air current.

The watt-hour meter was read after thermal equilibrium had been established, at half-hour intervals throughout the five-hour duration of the test, and at the end; while temperatures were measured six to eight times during the test. The warm room was maintained at approximately 70 F, and the cold room at some definite temperature between —30 F and 60 F, the coldest test being run first in any series of tests on any particular test specimen. Temperatures were measured at points opposite the center of each window light at a distance of one inch from the surface of the glass on both the warm and cold sides. Temperatures were also measured at a number of other points in the cold room, the guarded enclosure, and the warm room, at least one mercury thermometer being placed in each. Following each test, all the temperature readings of each thermocouple after equilibrium had been reached were averaged and the averages used in the computations.

Discussion of Results

The heat transmittance coefficients for plain glass, and glass as used in various window arrangements, are listed in Table 1, which includes a reference number, the authority, the heat transmittance coefficient, the glass set up (including the number of sheets of glass, air space thickness in the multiple-glazed cases, the total and test areas of the test panel, the description of the glass set up), the hot and cold side temperatures and velocities, and the mean temperature of the test.

An inspection of the table will show the heat transmittance coefficient values for single glass used as a single pane not glazed in a sash to be 1.02 (10)† and 0.89 (32) Btu per square foot per hour per degree Fahrenheit, as compared with a calculated value of 0.74 based on a film conductance coefficient for still air as given by Rowley.[7] Both of the values in this table were obtained under conditions nominally designated as normal air circulation. In the first case a fan was employed on the hot side and natural air circulation on the warm side; whereas in the second case, fans were used on both sides of the glass being tested. The exact velocities were not reported and may, to some extent, account for the difference in the heat transmittance values obtained. Further, a difference in methods employed to obtain the results may also account for some difference in the values resulting. The first method employs the large test room as a guard space; while the second method makes use of a guard box surrounding the hot box; the greater difficulty in maintaining constant temperature conditions in a large test room than in a small guard box is readily recognized as a possible source of error in the former method.

The heat transmittance coefficients for single-glazed windows in Btu per square foot per hour per degree Fahrenheit, with nominal still air conditions prevailing on both sides of the window, were found to be as follows: 0.64 (1) for a window of four 20 x 29 in. lights set in metal sash; 1.10 (2) for a window of four 30 x 30 in. lights set in metal sash; and 0.77 (22) for a window of nine 15¾ x 15¾ in. lights set in wood sash. The test areas of these windows were 16, 25, and 10.8 sq ft, respectively. The coefficients for single-glazed windows with the conditions of air motion being other than still[8] are as follows: 1.06 (4), (0 mph, 7 mph) for a window of 20 lights, each 12 x 18 in. set in wood sash; 1.19 (3), (0 mph, 7 mph) for the window of 20 lights set in solid rolled steel sash; 1.10 (5), (0 mph, 7 mph) for the window of 20 lights set in hollow metal sash; 1.26 (23),

†A parenthesized number designates the reference number in Table 1.

[7]Surface Conductances as Affected by Air Velocity, Temperature and Character of Surface, by F. B. Rowley, A. B. Algren and J. L. Blackshaw. (ASHVE Transactions, Vol. 36, 1930, p. 429.)

[8]The velocities in the parentheses represent those on the warm and cold sides of the window, respectively.

(0 mph, 5.6 mph) for the window of 9 lights set in wood sash; 0.99 (44), (2.18 mph, 0.66 mph) for the window of 9 lights set in steel sash. As is readily apparent, the results reported above vary over quite a range, being inconsistent with one another. It would seem that the variation in methods and velocities used would probably be responsible for this inconsistency; however, all attempts to reduce the above values to some common conditions of air movement did not result in more uniform values for the coefficients.

The heat transmittance coefficients in Btu per sq ft per hr per deg F for double glass having an air space of approximately ½ in., and with normal air circulation conditions on both sides of the window are 0.53 (12), 0.49 (13), and 0.50 (36), indicating that the difference in the methods employed in obtaining these results has but small effect upon the values. This is, of course, due to the fact that in double glazing the air space conductance, which remains constant for varying outside and inside air velocities is quite an important factor in the determination of the transmittance coefficient. Thus, variations in the air velocity on either side of a double-glazed arrangement for different testing methods will not greatly affect the results obtained, whereas, in the case of a single-glazed arrangement the heat transmittance coefficient is subject to any small variations in air velocity.

The magnitude of the effect of sash material and muntin shape and arrangement on heat flow through double-glazed windows is indicated by the results of tests (6), (8), and (9). These three tests were conducted in practically similar manner except for slight variations in air space and in the type and thickness of the glass used in the windows. However, since the air spaces used are either ⅝ in. or ¾ in. between which there is very little difference in conductance, and since the thickness of the glass is relatively unimportant in the final determinations of the transmittance coefficients, the results obtained present a good relative indication of the effect of sash material and muntin shape and arrangement on double-glazed arrangements. The values obtained in Btu per square foot per hour per degree Fahrenheit are as follows: 0.85 (6) for the solid steel sash window, 0.63 (9) for the hollow steel sash window, and 0.58 (8) for the wood sash window.

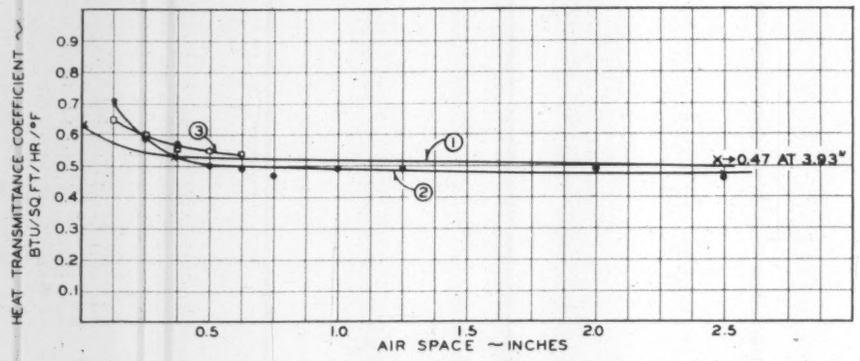


Fig. 4—Relation between heat transmittance coefficients in Btu/sq ft/hr/F and air space in inches, as determined by three investigators

1) Kreuger and Eriksson—plain glass; still air both sides
2) Rowley—plain glass; still air both sides
3) Pittsburgh Testing Laboratory—9 light steel sash window; hot side, 2.18 mph; cold side, 0.66 mph

The heat transmittance coefficients presented in (13), (14), and (15) are for double glass with a 1 cm (0.39 in.) air space, and square wood frame inserted between the glasses. In tests (14) and (15) there are, in addition to the frame, 4 horizontal and 4 vertical wooden rods, respectively, equally spaced through the air space between the glass lights. The heat transmittance coefficients given for tests (17), (18), and (19) are for arrangements similar to those of (13), (14), and (15) except that the air space in the latter tests was 10 cm (3.94 in.). The heat transmittance coefficients obtained for the first three arrangements, in Btu per square foot per hour per degree Fahrenheit, are 0.49, 0.51, and 0.51, respectively, indicating a very small increase in heat transfer through the double-glazed arrangement due to the addition of the wooden rods to the wooden frame. By calculation there is a greater heat transfer through a 1 cm thick wooden rod than through a 1 cm air space; therefore, the addition of the wooden rods should tend toward the increase shown. The heat transmittance coefficients obtained for the latter three arrangements with the 10 cm air space are all 0.45, indicating no change in the heat transfer caused by the addition of the wooden rods. Since a 10 cm thick wooden rod is a poorer heat transfer medium than a 10 cm air space, one would expect a decrease in the coefficient with the addition of the wooden rods. However, there is probably a convection effect set up between the lights of glass in the case of the 10 cm air space arrangement which tends to increase the heat transfer over that of the 1 cm air space arrangement. This increase as a result of convection opposes the decrease in heat transfer due to the addition of the wooden rods, and experimental evidence seems to show that these two opposite effects balance each other in the case of the 10 cm air space.

In Fig. 4 is seen the variation of heat transmittance coefficients, in Btu per square foot per hour per degree Fahrenheit, for double-glazed arrangements with air space in inches, as determined by three different investigators. Curves 1, 2 and 3, the results of data gathered by Kreuger and Eriksson, Rowley, and Pittsburgh Testing Laboratory, respectively, all indicate a decrease in heat transfer with increasing thickness of the air space. The rate of decrease is rapid for small air spaces, and decreases as the thickness of the air space increases so that a large part of the insulating effect is realized when the air space reaches ¼ in. A further small decrease in heat transfer is had for air spaces of from ¼ in. to ½ in. in thickness with little effect thereafter. This would indicate little advantage in air space greater than ½ in., while the practical problem of installing glass in sash seems to limit double-glazing to an air space of about ¼ in.

Curves 1 and 2 of Fig. 4 were

obtained for plain glass with still air conditions prevailing on both sides of the double-glazed arrangement, while curve 3 is for double-glazed steel sash with 2.18 and 0.66 mph velocities prevailing on the hot and cold sides, respectively. This latter curve shows the highest heat transmittance coefficients of the three curves.

Fig. 5 shows the variation of heat transmittance coefficients with increasing multiple-glazing and with a constant distance of 10 cm between the outermost panes. The curve indicates a decreasing heat transfer with increased glazing up to quadruple-glazing, showing that four thicknesses of glass in a particular space of 10 cm are the optimum number of lights one might employ.

Conclusions

The results of the several studies cited indicate transmittance coefficients varying over a wide range. These variations cannot be satisfactorily correlated with peculiarities in installation such as: The thickness and size of the glass lights; the number of sheets glazed in a single light; the material of which the sash is made; and the shape, size, and arrangement of the muntin strips. Nor can they be correlated with such test conditions as the actual air velocity over the surface, the method of heat input, the size of the panel tested, and the location and the method of temperature measurements.

As a result it may be concluded that the data now available are inadequate and inconclusive for practical usage. These studies do, however, serve as a basis for outlining a future and more comprehensive study of the entire subject. Such a future study should first include the acceptance of rigid test specifications governing the test arrangements mentioned previously, and should also include care in making such a study in order to take into account the effect of the various factors concerned with the glass installations in the window as mentioned before.

Such a set of test specifications which, in light of experience should be fashioned around the guarded hot box method, might well include the following:

1. The guard box, hot box, and cold box construction and assembly.

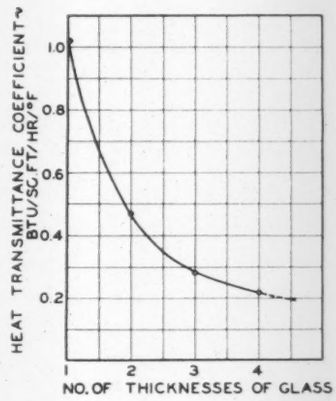


Fig. 5—Relation between heat transmittance coefficients in Btu/sq ft/hr/F and number of thicknesses of glass for multiple glazing

The distance between the outermost panes of glass is 10 cm (3.94 in.)

2. The preparation and placing of the test panel.

3. The heating and cooling equipment, including the type of equipment and the position of same in the respective boxes.

4. Temperature measurement, including the method to be used, the number and locations of the temperature measuring devices, and the accuracy to be obtained.

5. The mean temperature and temperature range of the tests.

6. The method of providing air movement and the place and method of measuring the same.

7. The method of measuring the heat supplied to the guard box.

8. A complete description of the window being tested.

PRE-SEASON GET-TOGETHER AT WASHINGTON, D. C.

The annual pre-season round-up of the Washington, D. C. Chapter of the Society was held at O'Donnell's Farm, Md., on September 17, and 35 members and guests enjoyed dinner with all the trimmings. An electric organ provided accompaniment for the usual barrack room ballads.

Following dinner all participated in a variety of games including horse racing, poker, dart throwing and barnyard golf. Private enterprise showed that it could still hold its head high in any company when William Hannigan turned in the most stage money at the conclusion of the festivities with Mr. Gates as runner-up, and both received suitable recognition of their skill.

I. M. Day, chairman of the entertainment committee, and G. R. Walz, who assisted him, were responsible for the success of the party.

Pres. F. A. Leser officially declared the affair adjourned at 10:30 p. m.

DEFENSE HOUSING

On October 14, the first anniversary of the government defense housing program authorized by the Lanham Act, Federal Works Administrator J. M. Carmody announced more than 100,000 defense homes completed, under construction or in the pre-construction stage.

The record shows: a total of 28,000 homes completed and already occupied or available for occupancy; approximately 64,000 more homes under construction; an additional 10,547 homes in the pre-construction stage.

The more than 100,000 homes provided for to date are divided among 371 defense housing construction projects in 41 States, the District of Columbia, Puerto Rico, Alaska, Hawaii and the Canal Zone.

Total estimated cost of the 371 projects is $397,847,000. Of the projects approved, 157 are to house the workers employed in defense industries and commercial shipyards. The remaining 214 projects are for army and navy enlisted and civilian personnel, of which 75 are for airports, 66 for army forts, camps and depots, 41 for navy yards and navy stations, and 32 for ordnance and powder plants, arsenals and proving grounds.

INDUSTRIAL HYGIENE FOUNDATION MEETING

The sixth annual meeting of the Industrial Hygiene Foundation will be held November 12 and 13, at Mellon Institute, Pittsburgh, Pa. The theme of the meeting will be "Defend the Defense Worker," according to an announcement of H. B. Meller, managing director. During the two-day conference 21 papers will be presented and among the questions to be discussed are the following:

How can industry reduce sick absences which claim millions of manhours yearly?

What are the most frequent health hazards? How can they be controlled?

What is the role of employee health in industrial relations?

What are the current legal trends, especially compensation for tuberculosis?

An Improved Method of Evaluating Dust Leakage from Air Filters

By L. K. Acheson* and H. A. Shumaker**, North Canton, Ohio

Introduction

DUST leakage from air filters has received much consideration in recent years and numerous methods of leakage evaluation have been proposed by various investigators. A quick and precise method of dust leakage evaluation that is designed to be as free from personal judgment as possible is described herewith. Outstanding features of the method are as follows:

1. Measurement of dust leakage obtained is that of total leakage of the filter rather than that obtained by sampling only a small portion of the dusty air stream.
2. The results are not affected by variations in dust concentration in the room air since the incoming air is 100 per cent cleaned by an electrostatic precipitator.
3. The human factor is almost entirely eliminated since the method does not involve weighing of dust leakage samples of extremely small magnitude.
4. Only downstream readings are required for the complete test of a filter.
5. The dust injector is sealed within the unit and therefore is not subject to errors due to leakage of atmospheric dust from the room air.

In particular, the test method and apparatus to be described were developed for testing vacuum cleaner bags which normally handle ordinary house dirt; however the procedure outlined is equally applicable for tests of any other type of dust filter. The method is based primarily upon the discoloration of a porous white Lintine target, the amount of discoloration being measured by photo-electric means. While the discoloration method has been used in the past by other investigators[1] the present method incorporates several innovations which to the authors' knowledge have heretofore not been used.

The discoloration method was chosen over the more commonly known ASHVE weight method because in this particular application it was known that the amounts of dust leakage to be measured would be extremely small and therefore any weight measurements would not only involve considerable time but also would be of questionable dependability. It was specifically desired that the method adopted, in addition to providing precise dust leakage measurements, should also be rapid, and, as nearly as possible, independent of the judgment of the operator.

Description of the Apparatus

A photograph of the test apparatus is shown in Fig. 1 and a schematic diagram is shown in Fig. 2. The test method consists of injecting a known quantity of dust of known particle size into a clean air stream which flows at a predetermined rate first through the filter to be tested and finally through a porous white Lintine target.[2] The entire leakage of the test filter is collected on the Lintine target.

Room Air Cleaned

Room air enters the apparatus at the left of Fig. 2, and passes through the electrostatic dust precipitator. The electrical precipitator, a Westinghouse Precipitron, Type S 375 cfm with 5/16 in. spaced Cor-Ten Cell and an 18 in. single cell ionizer, was made practically 100 per cent efficient in removing air suspended dust by operating at less than one-fifth rated air flow capacity. On leaving the electrostatic precipitator, the dust free air passes through centrifugal fan No. 1, and through the white check filter. If the electrostatic precipitator is operating satisfactorily, no discoloration of the check filter results. Motors driving the centrifugal fans of the apparatus were completely enclosed to prevent outside air from leaking into the ducts. After passing through the check filter a quantity of the air is by-passed to the dust injector.

[2]Lintine paper (1075 grams per yard).

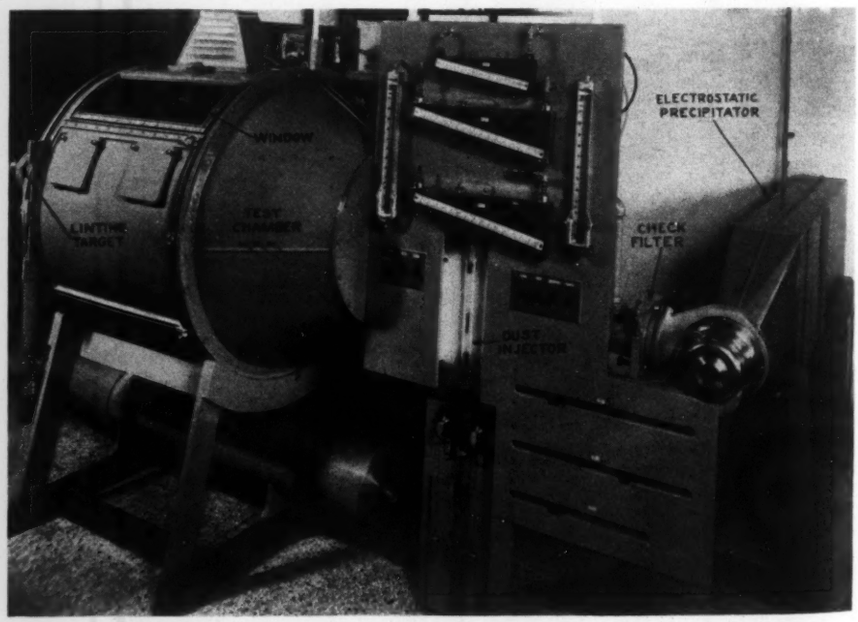


Fig. 1—General view of improved apparatus for measuring dust leakage from air filters

*Engineering Laboratories, The Hoover Co.
**Engineering Laboratories, The Hoover Co.
[1]A Test Method for Air Filters, by R. S. Dill. (ASHVE TRANSACTIONS, Vol. 44, 1938, p. 379.)

Dust Injector

An enlarged diagram of the dust injector system is shown in Fig. 3. The device is patterned after a dust feed unit developed by the U. S. Bureau of Mines;[3] however it differs from the device in that the incoming air is drawn from the electrostatically cleaned air stream of the main apparatus instead of from the room air. In this way there is no chance for dusty room air to leak in to the system through the injector and thereby introduce small errors in the results. Referring to Fig. 3, the dust injector unit draws clean air from the main duct, through an orifice which meters the flow and through the small centrifugal fan No. 2 which forces the air down the large glass tube (Pyrex 35 mm O.D.). The air in flowing down through the latter tube must pass through a stationary four-cut brass spiral element which produces a cyclone action in the air steam at the surface of the dust column. This cyclone action serves to pick up the dust particles as the dust column is mechanically raised at a uniform rate to approach the brass spiral, and the mixture of dust and air is then carried up through the small (15 mm O.D.) center glass tube and is exhausted into the main duct where it mixes with the clean air. Details of the elevating mechanism, which keeps the top of the dust column constantly at the proper level, are shown in Fig. 3. The dust sample, usually consisting of 100 grams, is contained in a cylindrical glass cup which is forced up inside the outer tubing by a piston which makes a seal with the wall of the outer tubing. The piston is lifted by a cord and cone pulley arrangement driven by self-starting synchronous clock motors. By means of the step cone pulley, the piston and dust column can be elevated at rates ranging from 0.5 in. to 4 in. per minute. For most tests a rate of travel of 2 in. per minute has been employed and found to be satisfactory.

[3]Testing and Design of Respiratory Protective Devices. (*U. S. Bureau of Mines Bulletin*, No. I. C. 7086.)

Test Chamber

Since this particular apparatus was primarily adapted to testing vacuum cleaner bags,[4] the dust laden air could be led directly into the mouth of the bag which was located within the test chamber (Fig. 2). In any case, the manner and velocity with which the dust laden air approaches the test filter should be similar to the conditions of service. Dust particles that escape from the test filter deposit on and serve to discolor the Lintine target. Photometer measurement of light transmission of the discolored Lintine target provides a numerical index of dust leakage of the test filter.

Measurement of Air Flow

After passing through the Lintine target, air is forced through fans Nos. 3-4, enters the surge chamber, and is exhausted through the streamlined orifice which meters the air flow for the test. The surge chamber between the exhaust of fan No. 4 and the orifice provides proper air flow conditions for precise measurement. For convenience all gages and controls were located on a single panel as shown in Fig. 1.

[4]For cases where it is desired to test filters in flat sample form a similar apparatus was built as shown in Fig. 6.

Electronic Photometer

A large number of phototube circuits have been published that could serve as a suitable means for measuring target discoloration. The following describes the application of but one of the many possibilities. The layout of the electronic photometer, used in measuring Lintine target discoloration, is shown in Fig. 4. The amount of light, emitted by the constant intensity light source, that passes through the target, striking the phototube, indicates the opacity or discoloration of the Lintine target. Naturally the more discolored the target, the smaller the quantity of light transmitted to the phototube. In order to facilitate handling, the Lintine target at the beginning of the test is placed in a hinged sheet metal frame having a circular cut-out the exact size of the large end of the photometer cone. The frame containing the target may thus be quickly inserted in either the dust test apparatus or in the photometer in much the same way as in loading plates in a camera. The photometer cone on the light source side is made with a reflecting interior surface which aids in diffusing the light evenly over the entire surface of the target. On the opposite or phototube side, a flat black is used on the interior surface of the cone. The phototube and amplifier unit used gave a nearly linear input *vs.* current output characteristic over the required range. Another feature of this unit was that the use of the amplifier resulted in a far greater sensitivity of dust measurement than that of any previous method tried. Assuming the phototube is dark (infinite resistance), the grid of the amplifier tube will

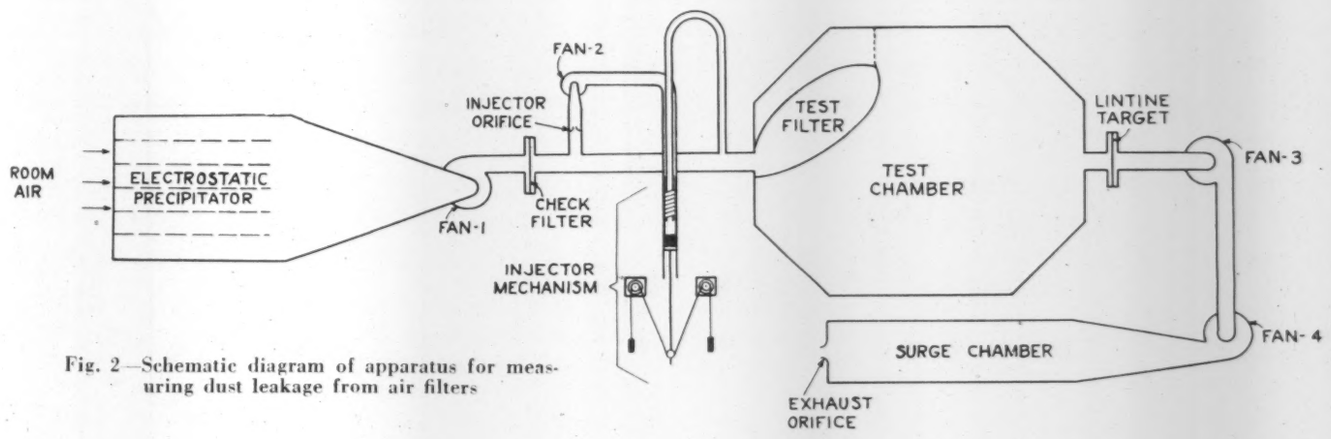


Fig. 2—Schematic diagram of apparatus for measuring dust leakage from air filters

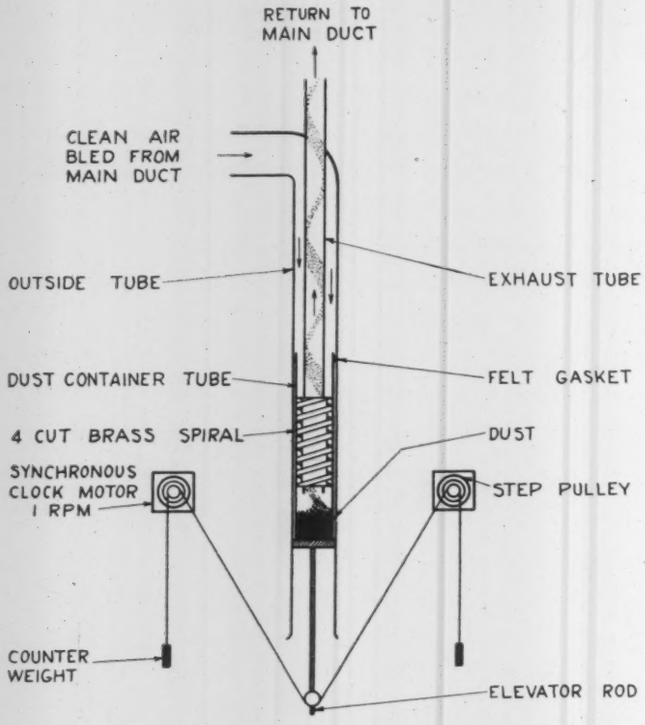


Fig. 3—Detail diagram of closed circuit dust injector

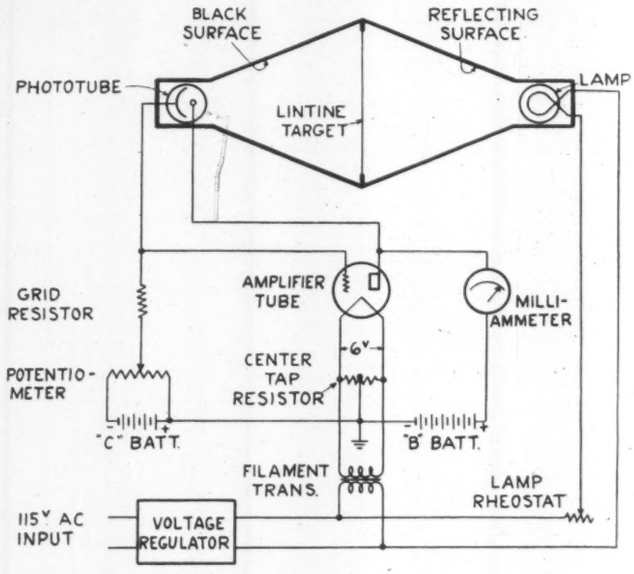


Fig. 4—Photometer (light transmission type) with wiring diagram for use in measuring target discoloration

be at a negative potential with respect to the filament, as determined by the potentiometer adjustment and C battery voltage. With this given grid potential a certain current will flow in the plate circuit. If the phototube is illuminated, the internal resistance drops to some finite value depending upon the intensity of the light, so that a current will flow under the influence of the B and C batteries, through the milliammeter, phototube and grid leak resistor. The grid leak voltage drop due to this current was such as to tend to make the grid of the amplifier tube less negative with respect to the filament. The decreased grid bias in turn increased the amplifier plate current. The amount of increase of plate current depended upon the bias which in turn depended upon the quantity of light reaching the phototube.

Since the light input *vs.* current output characteristic of the phototube was approximately linear over the range used during the tests, and since plate current values were only read from the linear portion of the grid voltage *vs.* plate current characteristic, the milliammeter readings of the photometer varied very closely linearly with the quantity of light transmitted through the Lintine target.

It was found that ordinary line voltage fluctuations of several volts out of 115, caused variations of the photometer lamp lumen output sufficient to cause photometer milliammeter fluctuations of as much as 20 per cent. A voltage regulator reducing line fluctuations to plus or minus 0.01 volt overcame this difficulty.

Selection of Standard Test Dust

Although vacuum cleaner bags normally handle house dirt, the use of the latter medium as a standard test dust was not permissible because no appreciable degree of uniformity was found to exist between any two batches of house dirt. As a result it was necessary to find a substitute.

Cottrell precipitate from the flue gases of a power plant with a percentage of white sand added was finally chosen as the standard test dust since this mixture had a density which corresponded reasonably well to that of house dirt and since it also possessed certain other desirable characteristics. For example, by proper fractioning and mixing it was possible to maintain a dust mixture which was fairly consistent throughout; there was a minimum tendency to agglomerate and thereby give an effective change in particle size; the dust mixture was fairly dark in color and had a large coefficient of light absorption; the mixture was not excessively hygroscopic, and its use did not involve any physiological hazards or danger from explosion. A sufficient supply of the Cottrell dust for a large number of tests was obtained from one of the power plants of the Potomac Electric Co., Washington, D. C.

The sand was added to the dust mixture partially to simulate the larger dust particles normally found in house dirt, and partially to serve as a scouring medium to remove any traces of the Cottrell dust that might temporarily adhere to the walls of the duct system of the dust injector apparatus. Composition of the test dust is shown in Table 1.

Table 1—Composition of Standard Test Dust

Material	Mesh Size	Composition Per Cent by Weight
Cottrell Precipitate	200-280	55
	280-pan	25
White Sand	65-100	10
	100-200	10
Total		100

Prior to selection of the Cottrell dust mixture as the standard test medium, other types of dust were tried including sieved house dirt, powdered coal, carbon black, powdered coke, and black iron oxide. For various reasons none of these were found to be as satisfactory as the Cottrell mixture.

Test Procedure

Dust for test purposes was thoroughly oven-dried before use. Rate of dust injection, chosen so as to be similar to test filter operating conditions, is controlled by the rate of elevating the dust container tube which is lifted by the cone pulley arrangement. The quantity of air bypassed through the injector is generally less than 5 cfm.

By maintaining a proper balance of the air load between centrifugal fans No. 1, and Nos. 3-4, the pressure within the test chamber can be controlled at will for any desired air flow. With the test filter in place and the apparatus ready to operate except for the Lintine target, the fans are turned on and the entire unit flushed with clean air to rid the ducts and test chamber of room air leaking into the various parts during setup.

The clean Lintine target is placed in the photometer before being mounted in the test apparatus. The grid bias is adjusted to give a fairly high plate current reading, as for example 10 milliamperes with this photometer. After flushing the test apparatus and after checking the clean Lintine target, it is put in place in the duct and the dust injector started. The difference between the clean and discolored target readings, at the same grid bias, represents the leakage of the test filter. An extra piece of Lintine on hand is useful in checking the constancy of the photometer readings.

With the circuit used test filters showing little dust leakage allow a relatively large amount of light to reach the phototube and hence cause large milliammeter readings. Conversely, test filters with considerable leakage cause low milliammeter readings. This was because the polarity of the grid voltage drop, regulated by the phototube, was such as to tend to make the grid of the amplifier tube less negative with respect to the filament. This reduction of grid bias results in increased plate current and consequently in higher milliammeter readings. For convenience the leakage index of the test filter was taken at the reciprocal of the milliammeter readings (multiplied by an arbitrary constant), so that larger indices corresponded to larger test filter leakages. It would have been possible to have large milliammeter readings correspond to relatively large test filter leakages and vice versa by arranging the circuit so that the polarity of the grid leak voltage drop would have been reversed.

In conducting filter tests using the new method, back pressure readings were, of course, taken simultaneously with each leakage reading and plotted for each filter. However, for sake of brevity, that phase of the work is not discussed in this paper.

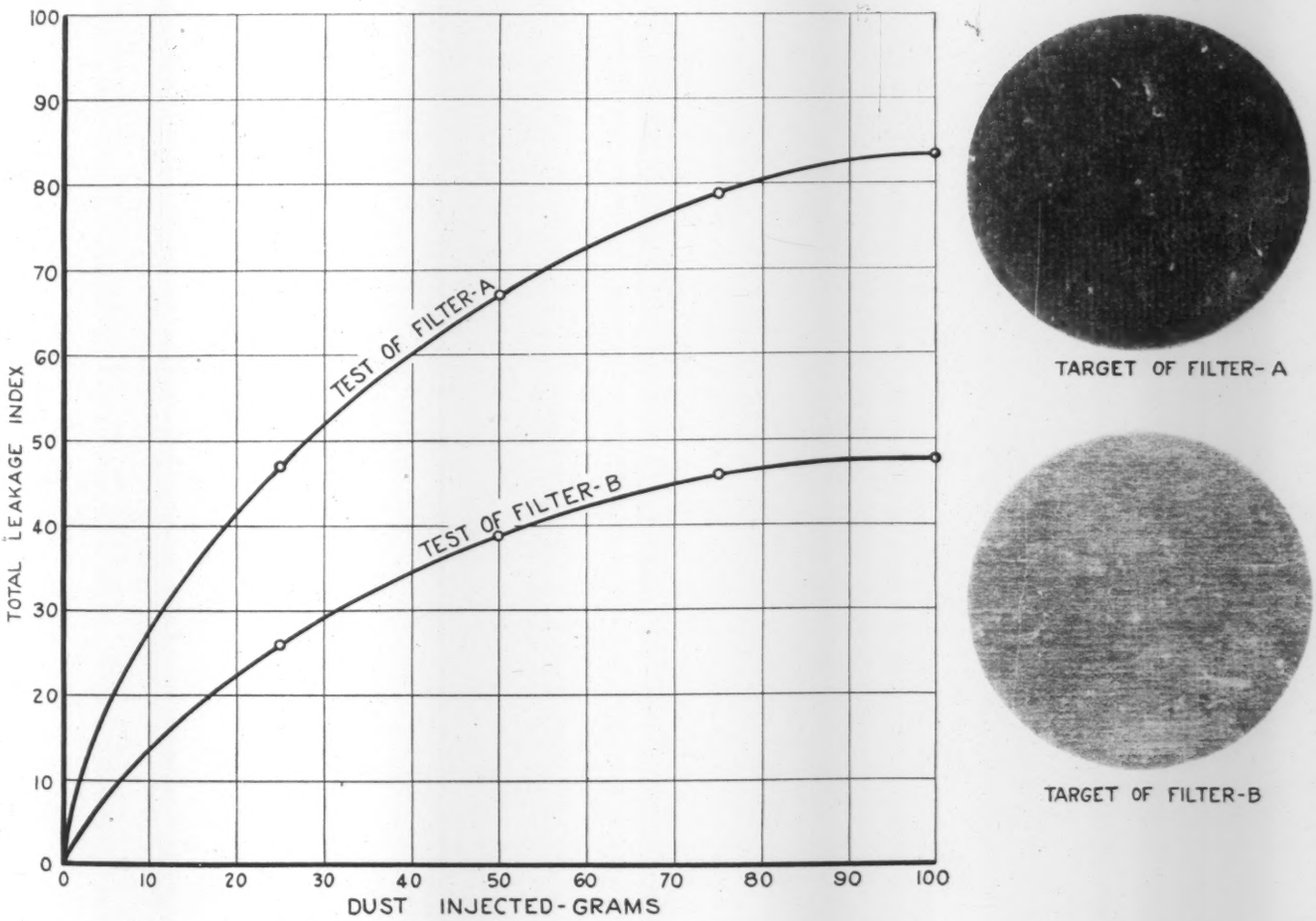


Fig. 5—Representative type of dust leakage curves plotted for various amounts of dust injected

Results

Results of tests made by the described method are normally plotted as shown by the representative curves in Fig. 5. This method of plotting gives a complete picture of the performance of the filter not only for one specific amount of dust but also for an entire range of dust quantities. This is a desirable feature since it has been found that the leakage properties of filters cannot always be determined by single readings. In fact a filter having the lowest leakage index at 25 grams injection may have the highest leakage at the 100 gram point.

Referring to Fig. 5, samples of the discolored Lintine are shown opposite each of the representative curves. Since the same piece of Lintine was used for each of the four points shown on the curve, the amount of dust shown on each target represents the total quantity that has leaked past the test filter.

Need for the electric precipitator intake cleaner was demonstrated early during the development when it was found that the daily variation in the amount of atmospheric dust in the room air was sufficient to cause large errors in the readings. In fact in some instances, it was found that without the precipitator in operation the discoloration from the room air alone was greater than that from the dust leakage of the filter. No difficulty has been experienced from that source since installation of the precipitator unit.

Reproducibility of Results

Because of the inherent variation in individual filters and since a given filter cannot be cleaned and thereby restored to its exact original condition, it has been difficult to obtain duplicate filters with which to check reproducibility of results. However, by taking consecutive filter samples from a given lot of filtering material a satisfactory degree of reproducibility has been obtained.

Correlation Between Cottrell Dust and House Dirt

While it has been stated that house dirt could not be used satisfactorily as a standard test dust because of its non-uniformity, still it was necessary to conduct a sufficient number of tests with house dirt to make sure that the Cottrell-sand mixture was a satisfactory substitute

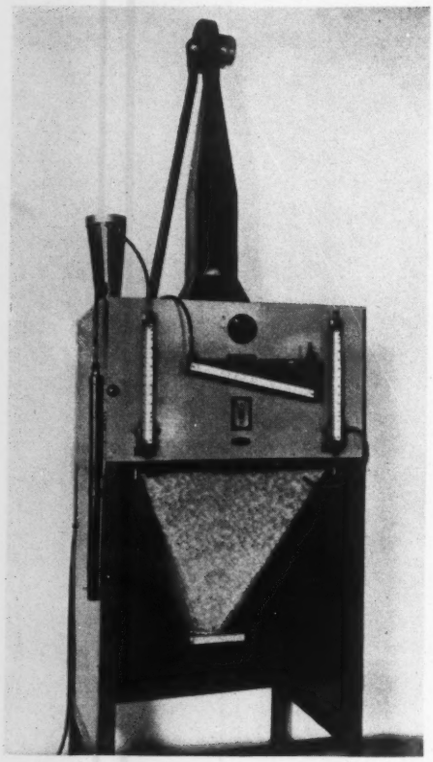


Fig. 6—Similar type of dust leakage apparatus adapted to tests of flat filter samples

and would give results comparable with those of house dirt. These tests demonstrated that there was a direct similarity in the results obtained from the two materials. The rate of leakage was accelerated in each case by the use of the Cottrell mixture (which was a desirable feature) but the ratio of leakage values obtained for various types of filters was the same with both types of dust.

Calibration of Unit

Although the photometric reading obtained from a discolored Lintine target gives an index of the amount of dust on the Lintine and therefore an index of the filter leakage, it should be pointed out that the relationship between the photometer reading and the dust quantity on the Lintine is not necessarily linear. In fact, since it is known that many of the dust particles in striking the Lintine may cover other dust particles already deposited, it is evident that linearity probably does not exist. This, however, does not detract materially from the method, but serves as a precaution not to assume that the leakage of one filter is twice that of another just because the leakage index of the one is numerically twice that of the other.

As work progresses, an attempt will be made to calibrate the unit by actually weighing the dust on the Lintine target for a given photometer reading. With chemical balances of proper sensitivity and with proper control of atmospheric conditions, this can undoubtedly be done. It is not a prime essential, however, to the successful use of the apparatus.

With regard to possible error in the dust leakage readings due to loss of some of the ultra fine dust particles through the Lintine, it has been found that by using the proper thickness of Lintine this loss is negligible.

Economic Value

In addition to the several features which have been discussed, the proposed test method has certain merits from an economic standpoint because of the saving in time required to make a test. For example, a complete test of a filter may be made in less than 30 min which is less time than that required for some of the even less precise methods.

NEW OFFICES FOR HATTIS

Robert E. Hattis, Chicago, Ill., consulting air conditioning, electrical, mechanical and power plant engineer, has moved his office to new and larger quarters in the Board of Trade Building. His many friends are invited to inspect his new quarters and enjoy the unusual view of the city.

KEEP LOCAL CHAPTERS COMMITTEE POSTED

E. K. Campbell, chairman of the Local Chapters Committee, invites any ASHVE member who plans a trip to one of the Chapter cities to advise the Local Chapters Committee of any such visit as far in advance as possible. He points out that the local chapter members enjoy meeting Society members from other cities and says that such visits will help promote a closer acquaintanceship within the membership of the Society and will also stimulate greater interest in local chapter activities.

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Philadelphia Meeting and Exposition

Benjamin Franklin Parkway from Philadelphia Art Museum

THE 4th week in January, 1942, offers busy days for members who attend the 48th Annual Meeting of the AMERICAN SOCIETY OF HEATING AND VENTILATING ENGINEERS and the *7th International Heating and Ventilating Exposition* in Philadelphia. The Philadelphia Chapter will be host for the occasion and an interesting program of sightseeing, inspection trips and entertainment has been planned.

For the sightseer Philadelphia has much to attract in its modern buildings, as well as in its historical points of interest. A bird's eye view of Philadelphia may be obtained from the observation tower of the Philadelphia Savings Fund Building. Looking South one sees the Philadelphia Navy Yard, the junction of the Delaware and Schuylkill Rivers, and a panorama of the vast oil refineries. Toward the East one views the Delaware River Bridge, and the city of Camden where the plants of RCA, Campbell Soup and New York Shipbuilding are located. On the North are the industrial areas of Frankford, Nicetown and Kensington, and looking West the observer will see the University of Pennsylvania campus, Convention Hall, and Fairmount Park, the largest in the world.

For those who are interested in renewing their acquaintance with the historical points, the center of attraction will be Independence Hall, in which the Constitution and Declaration of Independence of the United States were signed and which houses the Liberty Bell. Immediately adjoining is Congress Hall where the First Congress of the United States was held, and where Washington's second inaugural took place. Nearby is Carpenter Hall which was the scene of many important events of Colonial times.

The Betsy Ross House in which the first American flag was made, and Christ Church yard where Benjamin Franklin and several of the signers of the Declaration of Independence are buried will also be attractions. Among other fine old churches are Old Swede's Church, built in 1700, and St. George's Methodist Episcopal Church.

Of special interest to engineering visitors is Franklin Institute and Museum with its 4,000 exhibits, which dramatically demonstrate the part science plays in every day life and industry. Nearby is the Fels Planetarium, the Philadelphia Art Museum, the Rodin Museum, the Academy of Natural Sciences and the Public Library.

Among Philadelphia's educational institutions are the University of Pennsylvania, which recently celebrated its bi-centennial anniversary, Temple University, Girard College, founded for the education of orphaned boys, and Drexel Institute of Technology, which has a unique cooperative method of instruction.

There will be a series of social functions for both members and ladies, to be climaxed by the annual banquet on January 28 at the Bellevue-Stratford.

The technical sessions will be generally scheduled for the headquarters in the Bellevue-Stratford Hotel. Among the subjects selected by the Meetings Committee for discussion are: studies on panel heating and cooling performance, operating results of forced circulation hot water heating system, study of friction heads in screwed and welded elbows, mixing action of air streams, comfort with summer air conditioning, and thermodynamic properties of moist air.

Exposition Theme

The need for improving working conditions and speeding industrial production in plants engaged in National Defense operations lends special importance to the *7th International Heating and Ventilating Exposition*. Scheduled for January 26 to 30 next, in Philadelphia, the exposition will also stress the importance of greater efficiency and economy of operation, with a view to lower costs and fuel conservation in the heating, ventilating and air conditioning of all types of buildings.

Manufacturers of heating, ventilating and air conditioning equipment view the exposition as a meeting ground and forum for discussion and revision of plans in order to make adjustments for current contingencies and also to reveal the trend of planning for the post-war economy.

More than 250 exhibits are projected already for the Philadelphia Exposition, covering the entire field of heating, ventilating and air conditioning for industry, as well as human comfort in places of business and other institutions, and in homes. Firm names which comprise the bedrock of the industry constitute the principal listing on the roster, which also includes a number of newcomers.

As in the case of all previous heating and ventilating expositions, this one is under the management of the International Exposition Co., Grand Central Palace, New York.

C. B. Eastman

M. F. Blankin

H. B. Hedges

H. H. Erickson

W. P. Culbert

R. F. Hunger

E. H. Dafter

We Are Looking for You in '42.....

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H. H. Mather

A. C. Caldwell

R. D. Touton

Karl Rugart

J. H. Hucker

Mrs. M. F. Blankin

L. A. Childs

Chapter Officers for 1941-42

Northern Ohio

Paul Gayman
Secretary

J. A. Schurman, Jr.
President

W. M. Rowe
Treasurer

D. L. Taze

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BOARD OF GOVERNORS

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WESTERN NEW YORK HOLDS FIRST FALL MEETING

October 13, 1941. The first meeting of the 1941-42 season of the Western New York Chapter of the Society was held at the University Club. The minutes of the May meeting were read and approved.

At the request of Pres. W. R. Heath, B. C. Candee introduced his guest and new member, E. P. Heckel, Jr., and D. J. Mahoney introduced L. J. Jeselum as his guest.

The secretary read a letter from V. J. Jenkinson, Toronto, in which the Ontario Chapter requested the Western New York Chapter to consider the possibility of holding a joint meeting of the Buffalo and Ontario Chapters at some intermediate point, such as Niagara Falls. The letter further stated that the Ontario members felt such a meeting would be highly beneficial in fostering mutual interests and creating a bond of fellowship and understanding between the two chapters, and at the same time, help in creating and fostering good will between the neighboring countries. President Heath asked for comments from the floor, and Mr. Candee thought that the suggestion had merits and offered the opinion that it was a matter to be left up to the officers of the Chapter for decision. Roswell Farnham spoke in a similar manner and S. W. Strouse offered some elaboration in that he suggested that this might also offer an opportunity to invite the ASHVE members from Rochester and vicinity to join the Toronto and Buffalo groups. He further suggested that it might be possible to arrange a tour of inspection through either the Bell Aircraft or Curtiss-Wright Aircraft Plants and then adjourn to the General Brock Hotel for dinner and the joint meeting. The consensus was that this matter should be referred to the officers for a decision.

Following the treasurer's report, President Heath introduced F. C. McIntosh, Pittsburgh, Pa., as the speaker of the evening, stating that he had long been active in Society affairs, having served on the Council and is now a member of the Committee on Research. Mr. McIntosh spoke on the subject "Our

Research Committee—Asset or Liability," which he delivered in an interesting manner, illustrating his remarks with lantern slides showing various phases of the projects under consideration by the Society. His talk, according to Herman Seelbach, Jr., secretary of the Chapter, was very well received by the members and guests. Following a discussion the meeting adjourned.

SOCIETY RESEARCH TOPIC AT ST. LOUIS

October 2, 1941. The regular monthly meeting of the St. Louis Chapter was held at the DeSoto Hotel, with Pres. D. J. Fagin presiding. The secretary read the minutes of the September meeting, which were approved as read.

Following the report of the treasurer, B. C. Simons announced that plans were almost complete for the November meeting which would be addressed by an FBI representative on the subject of Mechanical Devices used in Crime Detection.

With no further business before the meeting, a recess was taken, and the members and guests went to the Engineers' Club of St. Louis, where President Cooper of the Engineers' Club turned the annual joint meeting of the Club and the Chapter over to President Fagin. President Fagin introduced the guest of the evening, John James, New York, technical secretary of the ASHVE.

Mr. James delivered an illustrated talk on the research program of the Society, tracing its history and development into the most comprehensive program of research known to be underwritten by any professional society. He described the cooperative arrangements between the Committee on Research and the Universities and Institutions throughout the country, and covered the need for the various major classifications of research, the methods of investigation, and the dissemination of the results. He gave life and interest to a difficult subject, and made it clear why and how the Society's funds and efforts were so comprehensively employed in such a program.

There was considerable interest evidenced by both the Chapter members and the Club members and guests, and discussions took a great deal of time after the formal presentation was finished. A rising vote of thanks was extended Mr. James, and the meeting adjourned, according to J. H. Carter, secretary of the Chapter.

September 9, 1941. The first meeting of the season held by the St. Louis Chapter was at the DeSoto Hotel, St. Louis, and in the absence of Pres. D. J. Fagin, the vice-president, M. F. Carlock presided. The attendance was reported as 26 members and guests.

Following the reading of the minutes of the May meeting, which were approved as read, the treasurer gave his report. The chairman of the membership committee, B. L. Evans, reported on new applications, followed by a motion, which was duly seconded and carried, that the regular chapter meetings for the balance of the chapter year be held at the DeSoto Hotel.

Some of the members expressed their opinion that these meetings should start at 6:00 p. m. instead of a half hour later, and this was left to the meetings committee to determine by the next meeting.

B. C. Simons, chairman of the meetings committee, introduced G. W. F. Myers, St. Louis, who presented a complete and able discussion of the recent paper on air raid shelters by W. A. Bechtler of Zurich, Switzerland, which was published in the June 1941 Journal of the Society. Mr. Myers also referred to the recent publication, Civil Air Defense, by Lt. Col. A. M. Prentiss, General Staff, U. S. Army. He described the bomb-proof requirements of a shelter, penetration of a typical 600 lb bomb dropped from 25,000 ft (985 fps velocity, penetration 6-6 in. reinforced concrete floor slabs), and the ventilation and filtration requirements, sanitary and other needs of typical shelters. Mr. Myers also referred to the material presented by W. L. Fleisher, president of the ASHVE.

After a brief intermission J. H. Carter presented a discussion on the subject of frozen foods and locker plants, covering the history of such plants, their use by either farm or city families, problems of design, engineering and operation.

The meeting adjourned at 10:00 p. m., according to J. H. Carter, secretary of the Chapter.

DR. GIESECKE VISITS SOUTHERN CALIFORNIA

October 8, 1941. The first meeting of the 1941-42 season was held at the Stevens Nikabob, Los Angeles, and according to the secretary's report, there were 44 members and guests in attendance.

After dinner a short business session was held, and it was voted to change the meeting night from the second to the first Wednesday of each month, and the meeting place changed to Eaton's Chicken House. It was announced that a roster of the membership combined with a directory of contractors, manufacturers and distributors of materials and equipment relating to the heating, ventilating and air conditioning industry within the jurisdiction of the Southern California Chapter, would be published in December.

The business session was followed by a most interesting talk by Dr. F. E. Giesecke, past president of the ASHVE. His subject "How to Use Research Results in Engineering Practice" illustrated with slides, demonstrated very clearly the necessity and value of research to the industry. He also brought out the fact that every member of the Society was contributing to this research as the major portion of the annual dues are appropriated to maintain the Research Laboratory in Pittsburgh to carry on with research already started. A unanimous vote of thanks was given to Dr. Giesecke, and the meeting adjourned.

ONTARIO WELCOMES MAJOR FOX

October 6, 1941. The first regular meeting of the season was held at the Royal York Hotel, with 85 members and guests in attendance. Following dinner, A. M. Dion put on a comical act mimicking Adolph Hitler.

President Tasker then called upon the principal speaker of the evening, Major J. H. Fox, a member of the Chapter, who is now with the R. C. O. C., 5th Division Field Army Workshop at Camp Borden. Major Fox outlined his activities in the Army since joining the Canadian Forces last year, and told of his trip to England, and of conditions as they now exist there, especially in

London. Major Fox was among the few Canadians who went to France following Dunkirk. He also told of his return trip to Canada where he has been an instructor at Camp Borden, Ontario.

Following Major Fox's address, H. D. Henion extended a vote of thanks for what the Chapter hailed as one of their most interesting meetings.

Major Fox advised the Chapter that another one of the Chapter members, J. W. Bishop has now been promoted to Major, and is with the Armored Division of the Canadian Corps.

V. J. Jenkinson announced that Dr. F. C. Houghten, director of the Society Research Laboratory, Pittsburgh, Pa., would be the guest speaker at the November meeting.

RESIDENTIAL HEATING OPENS ATLANTA'S SEASON

September 9, 1941. The first regular Fall meeting of the Atlanta Chapter was held at the Atlanta Biltmore Hotel, and was called to order by Pres. S. W. Boyd with an attendance of 29 members and guests.

The minutes of the previous meeting were read and approved, and President Boyd then reported on various items of general interest to the Chapter. He called attention to a visit from the secretary of the Society, A. V. Hutchinson, during the summer. Mr. Hutchinson met with the officers and committee chairmen of the Chapter and discussed matters of interest to the Society, with particular reference to membership and an effort to plan chapter programs well in advance.

President Boyd also reported that a memo from the headquarters office advised that the *American Society of Engineers* was recently investigated by the U. S. postal authorities and found to be the pet idea of one man and apparently simply a means of income for him.

Attention was brought to the letter recently sent out by the chairman of the program committee, M. M. Crout, which listed the subjects and speakers for the meetings for the coming year.

Announcement was then made that the regular Monday Engineers luncheon would be sponsored by the *Georgia Engineers Society* this year.

Mr. Crout reported briefly on his committee's work during the summer months which resulted in the program being set up for the year. He stated that the subjects chosen were a choice of the majority and that these programs should therefore result in better attendance at the meetings. He further advised that the poll that was taken on the topics to be discussed at the monthly meetings resulted in some very interesting facts, which would be tabulated and distributed to the membership. President Boyd complimented Mr. Crout and his committee on their work in preparing the program, and stated that, to his knowledge, this was the first time the chapter's program had been set up a year in advance.

T. T. Tucker, chairman of the research committee, gave a report on the results to date of the attic fan test that is being sponsored by the chapter, and presented a comparison between the two test houses, one with and one without an attic fan. He stated that the committee hoped to finish the test work before cool weather set in and that the data would then be compiled so that Georgia Tech can submit a paper on this test for the 48th Annual Meeting of the Society in Philadelphia in January.

T. T. Tucker also gave a brief report on the activities of the *Georgia Engineering Society* stating that a complete program had been arranged for the coming year and that their activities would officially begin at the first meeting in October.

C. L. Templin, chairman of the membership committee, reported on the new applicants, and requested that the members of the chapter continue to cooperate with the committee in order to bring into the Society any qualified engineers whom they feel would be an asset to the Chapter.

L. F. Kent announced that he would endeavor to give some publicity to the program scheduled, and also to the attic fan test.

Following reports on codes and the finances of the chapter, President Boyd introduced the speaker of the evening, M. W. Wise, heating and ventilating contractor in Atlanta, who gave an interesting talk on the subject of residential heating using modern equipment. Mr. Wise traced the evolution of residential heating over the past 20 years, pointing out the types of heating systems that were popular at different times, and how with the help of the engineer some of these systems were brought back into popularity years after they were considered outmoded. He also pointed out that in his opinion residential heating demanded the best efforts of the engineer and contractor because in residential heating it is necessary to satisfy the rigid requirements set forth by architects and home owners. In conclusion the speaker stated that he felt that a great deal of work is yet to be done in the field of residential heating and that considerable improvement will be made in the heating equipment which is now considered modern.

At the conclusion of his talk, Mr. Wise answered numerous questions that were asked, and judging by the interest shown in this talk, it is evident that the Chapter's decision to have the subjects at the monthly meetings handled by members, wherever possible, was a good one.

President Boyd then announced that the next meeting would be in October, at which time John James, technical secretary of the Society, would speak on Recent Research Trends in Heating, Ventilating, and Air Conditioning.

As reported by the secretary, A. H. Koch, the meeting adjourned at 9:15 p. m.

FIRST FALL MEETING AT PITTSBURGH

October 14, 1941. Pres. E. C. Smyers called the first meeting of the season to order in the Conference Room of the Pittsburgh Y. M. C. A. The minutes of the May meeting were read and approved as read.

P. C. Strauch, chairman of the membership committee, briefly discussed the proposed membership roster, the printing of which had been approved at the executive committee meeting held on September 30, 1941.

There being no old or new business, President Smyers introduced S. H. Downs, chief engineer of the Clarage Fan Co., Kalamazoo, Mich., who was brought to Pittsburgh under the auspices of the Chapter Re-

lations Committee. Mr. Downs spoke on the research activities of the Society in accordance with a nation wide plan of the Committee and the Council.

Following his talk on research, Mr. Downs discussed informally the application of fans to heating and ventilating work. A lively discussion followed, and the meeting adjourned with a rising vote of thanks to Mr. Downs. There were 37 members and guests in attendance.

MINNESOTA ACTIVITIES

September 6, 1941. More than 70 members and guests were on hand for the first meeting of the season, which was held at the Hotel Stoddard, La Crosse, Wis. During the day the members from Minnesota, Iowa and Wisconsin were guests of the Trane Co., and a feature of the day was a tour of the factories in La Crosse.

At the banquet in the evening, F. O. Jordan, Detroit, Mich., representative of Albert Kahn and Associates, told his audience that wars are looked upon as great tradition breakers in the building construction and equipment field. He went on to say that tradition can be a danger because if it crystallizes too far it eliminates improvement and advancement. He also stated that war forces the adoption of new ideas, and in the past wars were fought on the fringes of nations by human beings, with death and destruction not extending to the interior. Mr. Jordan said, "Now wars are fought by machines of long range. The nation with the most and best machines probably will win. For that reason it is essential to keep production at a high peak."

Mr. Jordan predicted a rapid growth in air conditioning after the war and pointed out that the national defense program and the war are emphasizing the importance of air conditioning in industrial plants.

N. D. Adams, acting as master of ceremonies during the banquet, introduced Messrs. Arthur Holmes, D. C. Minard, and Norbert Downey of The Trane Co., and the Messrs. T. R. Johnson, Des Moines, Ia., from the Iowa Chapter, F. W. Goldsmith, Milwaukee, Wis., representing the Wisconsin Chapter, and H. M. Betts, president of the Minnesota Chapter of the Society, and Marshall Taft of the *Minnesota Professional Engineers Society.*

Have You Registered?

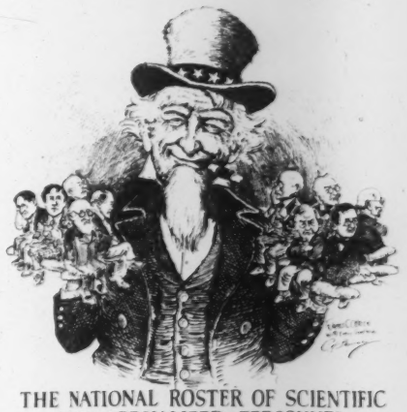
By T. H. Urdahl, *Chairman*
Membership Committee

Each member of the ASHVE should recognize the manifold advantages of his membership in the Society, which today provides a means of supplying the Government with sources of specialized knowledge for effective application in the defense program at a critical moment. A member may be just the right man for an important assignment and membership in the Society may be the means of bringing his qualifications to the attention of many agencies through the National Roster of Scientific and Specialized Personnel.

The National Roster is jointly administered by the National Resources Planning Board and the United States Civil Service Commission, and it offers Government agencies and others an outline of the potential abilities of men in many specialized fields of engineering.

Today first consideration should be given to the following:

1. If you have not already filed your name and qualifications with the National Roster, do so at once.
2. Suggest to your fellow workers that they register.
3. Encourage Society membership among those who are engaged in heating and ventilating or allied fields, but who are not now classified in this group.
4. The Society, as an organization as well as through individual members, is already making large contributions to the defense program,



THE NATIONAL ROSTER OF SCIENTIFIC AND SPECIALIZED PERSONNEL

and this assistance can be maintained and increased through your own personal service.

5. The Government needs the service of many engineers, but it should be remembered that it does not want to take you away from your job or your job away from you. It does want your help temporarily, or the help of experienced engineers in your organization whom you might "loan" for a period of days, weeks or months.

The following excerpts from the National Roster indicate that the potential abilities of the ASHVE membership are by no means completely listed and there is an urgent need for a complete listing of members of the engineering profession specializing in heating, ventilating and air conditioning:

HEATING, VENTILATING, REFRIGERATING AND AIR CONDITIONING ENGINEERING (and related fields)

HEATING
Domestic
——Gas 58.0.02
——Oil 58.0.04
——Coal 58.0.06
Industrial
——Gas 58.0.12
——Oil 58.0.14
——Coal 58.0.16
——Power plants .. 58.0.20
——District 58.0.30

VENTILATION
——Fumes 58.0.50
——Bacteria 58.0.60
——Gases 58.0.70
——Dust 58.0.80
——Industrial exhaust 58.0.90

AIR CONDITIONING (Transportation)
——Railroad 58.1.00
——Ship 58.1.10
——Airplane 58.1.20
——Bus 58.1.30
——Auto 58.1.40

AIR CONDITIONING (n.e.c.)
——Industrial, general 58.1.50
——Domestic, comfort 58.1.82
——Drying systems. 58.1.85
——Vapor recovery. 58.1.90
——Treatment of disease 58.1.95

REFRIGERATING MACHINERY (Commercial) (design, development, etc.)
——Condensing units 58.2.00
——Controls 58.2.10
——Evaporators ... 58.2.20
——Self-contained fixtures (ice cream cabinets, water coolers, soda fountains, freezing cabinets, etc. 58.2.30
——Compressors ... 58.2.40
——Brine coolers... 58.2.50
——Absorption systems 58.2.60
——Lubrication 58.2.70
——Fluid flow..... 58.2.80

REFRIGERATION (Transit)
——Freight cars.... 58.3.00
——Refrigerated truck bodies.. 58.3.10
——Refrigerated ships 58.3.20

REFRIGERATION APPLICATIONS (n.e.c.)
——Cold storage.... 58.3.50
——Dry ice manufacture and service 58.3.60
——Freezing, quick. 58.3.70
——Ice manufacture and service... 58.3.80

PLUMBING
——Water supply and drainage. 58.4.00
——Vacuum cleaning 58.4.10
——Sprinkler systems 58.4.20
——Compressed air. 58.4.30
——Cafeteria and restaurant layouts 58.4.40

ELEVATORS
——Passenger 58.4.50
——Conveyors 58.4.80

RESEARCH
——Physiological .. 58.5.00
——Treatment of disease ... 58.5.05
——Air purification. 58.5.10
——Heat transfer.. 58.5.20
——Flow of fluids.. 58.5.25
——Corrosion 58.5.30
——Instruments ... 58.5.35
——Insulation 58.5.40
——Psychrometry .. 58.5.45
——Fuels and combustion .. 58.5.50
——Sound control.. 58.5.55

Function

——Research and development . 0.
——Design and manufacture . 1.
——Editors and writers ..... 2.
——Consultants .... 3.
——Installation, erection and operation .... 4.
——Professors and instructors .. 5.
——Administration . 6.
——Specifications and testing... 7.
——Application .... 8.
——Other ..... 9.